British Chemical Engineering

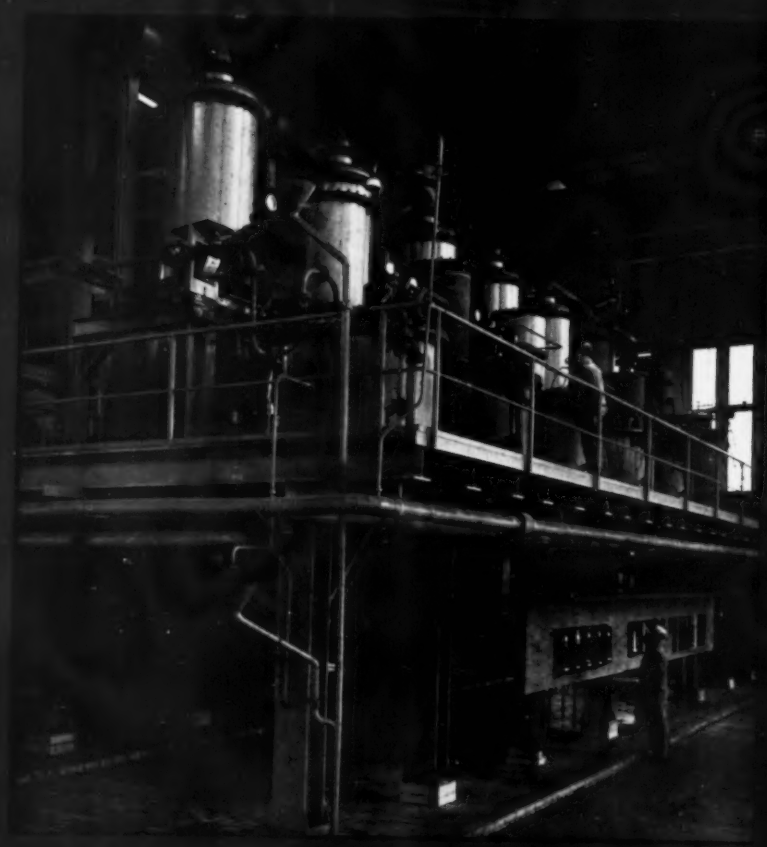
January 1960 Price 2s. 6d.

NOVA SCOTIA TECHNICAL COLLEGE HALIFAX N. S.

-temperature Heat Exchange Symposium

Applications of Regenerators and Heat Exchangers (page 15); Design (page 18) and Overall Economics
Plate-fin Heat Exchangers (page 40); and Secondary Surface Heat Exchangers (page 27)

BCE 6623 for further information



ACID CONCENTRATION

This picture shows a recently completed Submerged Combustion Plant for the concentration of Sulphuric Acid used in the manufacture of Isopropyl Alcohol and secondary Butyl Alcohol. The evaporative capacity is approximately 5 tons per hour, and the plant is now being enlarged to double capacity.

This equipment was installed for Messrs. Rheinpreussen A.G. für Bergbau und Chemie, Homberg, Niederrhein, Germany.

Other plants have been supplied or are in course of construction or erection for U.S.S.R., France, Holland, Belgium, South Africa, India and the U.S.A.

NORDAC

CHEMICAL ENGINEERS

MEMBERS OF THE WOODALL-DUCKHAM GROUP OF COMPANIES

A HEYWOOD PUBLICATION

British Chemical Engineering

THE JOURNAL DEVOTED TO THE TECHNIQUES OF CHEMICAL ENGINEERING APPLIED TO INDUSTRY

JANUARY 1960
Vol. 5 No. 1

Editor:
J. O. S. MACDONALD, B.Sc., A.R.I.C., A.M.I.Chem.E.

Associate Editor:
I. BERKOVITCH, Ph.D., F.R.I.C., A.M.I.Chem.E.

Assistant Editor:
WILLIAM D. CLIFFORD

A multi-tubular low-temperature heat exchanger in course of construction at the Edmonton works of British Oxygen Engineering Co. Ltd.

Classified Advertisements, see page 8

BRITISH CHEMICAL ENGINEERING is published monthly by HEYWOOD & CO. LTD., Drury House, Russell Street, Drury Lane, London, W.C.2 (Tel.: TEMple 3422), on the 20th of the preceding month. Annual subscription £2 U.K., £2 10s. 0d. sterling overseas for 12 issues post free ($7.50 U.S.A. and Canada) or by standing order through newsagents at 2s. 6d. a copy net. BRANCH OFFICE: Royal Buildings, 2 Mosley Street, Manchester, 2. Tel.: Central 9935. © Heywood & Co. Ltd., 1960. Must not be reproduced without written permission.

MAIN ARTICLES

Low-temperature Heat Exchangers *by P. M. Schuftan, Ph.D., M.I.Chem.E.* 12
Les échangeurs de chaleur à basse température
Tieftemperaturwärmeaustauscher
Низкотемпературные теплообменники

The Application of Regenerators and Heat Exchangers *by D. E. Ward, Ph.D., A.M.I.Chem.E.* 15
L'application des régénérateurs et des échangeurs de chaleur reversibles
Die Anwendung von Regeneratoren und Reversierwärmeaustauschern
Применение регенераторов и реверсивных теплообменников

Applications and Design of Plate-fin Heat Exchangers
by W. H. Denton, B.Sc., F.Inst.P., and D. E. Ward, B.Sc., Ph.D., A.M.I.Chem.E. 18
Les applications et la construction des échangeurs de chaleur à plateaux joints par des nervures
Die Anwendung und die Ausführung von Wärmeaustauschern mit rippenverbundenen Platten
Применения и конструкция пластинчато-ребристых теплообменников

Secondary Surface Heat Exchangers *by A. G. Lenfesty, B.A.* 27
Les échangeurs de chaleur à surfaces secondaires
Wärmeaustauscher mit Sekundäroberflächen
Теплообменники с вторичными поверхностями

A Simplified Regenerator Theory *by W. F. Schalkwijk, Ph.D.* 33
Une théorie simplifiée sur les régénérateurs
Eine vereinfachte Regeneratortheorie
Упрощенная теория регенераторов

General Considerations on Efficiency and Costs of Regenerators *by H. London* 35
Etude générale du rendement et des coûts des régénérateurs
Allgemeine Betrachtungen über die Leistung und Kosten von Regeneratoren
Общие соображения о к.п.д. и стоимости регенераторов

Heat Transfer Accompanying the Condensation of Mixed Vapours
by G. G. Haselden, Ph.D., and W. A. Platt, M.Sc. 37
Transfert de chaleur accompagnant la condensation de vapeurs mixtes
Die der Kondensation von gemischten Dämpfen beifolgende Wärmeübergabe
Теплопередача, сопровождающая конденсацию смешанных паров

Overall Economics of Plate-fin Heat Exchangers *by W. H. Denton, B.Sc., F.Inst.P.* 40
L'économie générale des échangeurs de chaleur à plateaux joints par des nervures
Allgemeine Wirtschaftlichkeit von Wärmeaustauschern mit rippenverbundenen Platten
Общая экономика пластинчато-ребристых теплообменников

OTHER FEATURES

Lender: Low-temperature Heat Transmission 11
Reddish Chemical's New Factory 17
Water Treatment Plant for Fords 26
Operating and Testing of Connersville Motors 32
American Letter: Activities in the U.S. Fertiliser Industry 50
Nomogram: Physical Properties of Hydrogen at Atmospheric Pressure, by A. M. P. Tans 52
Letters to the Editor 56
French Industries Cut Relining Costs 58

REGULAR FEATURES

Digest 1
Process Engineer's Notebook 54
Book Reviews 60
World of Chemicals 62
Plant Equipment News 64
Month's News in Brief 74

Résumés

B.C.E., 5, 1, 12. U.D.C. No. 66.045:1-974

LES ÉCHANGEURS DE CHALEUR À BASSE TEMPÉRATURE

par P. M. Shuftan

Une revue d'ensemble des différents genres d'appareils de transfert de chaleur utilisés dans les installations de séparation des gaz à basse température, le régime de leur exploitation et les buts poursuivis dans leur construction. Les effets de la contamination, l'importance des différences de température reduites au côté chaud (point d'entrée de l'air comprimé et de sortie de l'azote), des pertes de charge basses et des differences de température peu importantes dans les rechaufferurs sont des facteurs d'importance particulière dans l'économie des procédés à basse température.

B.C.E., 5, 1, 15. U.D.C. No. 662. 995-974: 66.045. 1-974

L'APPLICATION DES RÉGÉNÉRATEURS ET DES ÉCHANGEURS DE CHALEUR REVERSIBLES

par D. E. Ward

Les échangeurs de chaleur à basse temperature doivent être auto-nettoyables, compacts, peu coûteux et capables de fonctionnement avec des différences de température très reduits. Les avantages relatifs des échangeurs reversibles et des régénérateurs pour les tâches de transfert de chaleur à basse température sont comparés.

B.C.E., 5, 1, 18. U.D.C. No. 66.045.1:66.074.8

LES APPLICATION ET LA CONSTRUCTION DES ÉCHANGEURS DE CHALEUR À PLATEAUX JOINTS PAR DES NERVURES

par W. H. Denton et D. E. Ward

Les avantages de l'échangeur de chaleur à plateaux joints par des nervures pour le transfert de chaleur et l'épuration des gaz dans les séparateurs de gaz à basse température sont discutés. Des méthodes sont décrites, avec quelques exemples, pour la construction de ces agrégats pour des tâches spécifiques de transfert de chaleur et de perte de charge, ainsi que pour l'estimation des effets de mauvaise distribution d'écoulement.

B.C.E., 5, 1, 27. U.D.C. No. 66.045.1

LES ÉCHANGEURS DE CHALEUR À SURFACES SECONDAIRES

par A. G. Lenfestey

La construction des échangeurs de chaleur à plateaux ondulés et les possibilités d'aménagement de ce type de surface sont décrits, ainsi que les applications de ces derniers aux procédés de basse température et leurs caracteristiques de transfert de chaleur.

B.C.E., 5, 1, 33. U.D.C. No. 662.995

UNE THÉORIE SIMPLIFIÉE SUR LES RÉGÉNÉRATEURS

par W. F. Schalkwijk

L'auteur dérive une formule simple pour le calcul du rendement d'un régénérateur, basée sur le fait qu'en pratique la plupart des régénérateurs possèdent un champ de température linéaire dans leur centre.

B.C.E., 5, 1, 35. U.D.C. No. 662.995:66.012.7

ETUDE GÉNÉRALE DU RENDEMENT ET DES COUTS DES RÉGÉNÉRATEURS

par H. London

L'importance des facteurs différents agissant sur la construction des régénérateurs, notamment perte de charge, distribution d'écoulement, conduction de chaleur longitudinale et retention des gaz, est discutée. On dérive une serie de formules fournissant un moyen de construire les régénérateurs à un coût global minimum.

B.C.E., 5, 1. 37. U.D.C. No. 66.045.2:66.048.2

TRANSFERT DE CHALEUR ACCOMPAGNANT LA CONDENSATION DE VAPEURS MIXTES

par G. G. Haselden et W. A. Platt

Une critique des méthodes actuels pour le calcul des coefficients de couches de condensation pour les différences de température reduites. Les éléments de construction requises d'un condensateur approprié pour les charges de basse temperature sont indiqués et une construction efficace est décrite.

B.C.E., 5, 40. U.D.C. No. 66.045.1:66.074.8: 66.012.7

L'ÉCONOMIE GÉNÉRALE DES ÉCHANGEURS DE CHALEUR À PLATEAUX JOINTS PAR DES NERVURES

par W. H. Denton

Les frais de consommation d'énergie et de premier établissement des échangeurs de chaleur constituent une partie considerable des frais globaux des procédés se deroulant a moins de 100° K environ. Les installations de distillation à l'air et au hydrogène en fournissent des exemples. A l'aide d'exemples de construction optimum de ces échangeurs, l'auteur démontre que les frais d'énergie de refroidissement, de pertes de charge et de premier établissement respectivement restent toujours dans une relation de 3 : 1 : 2.

Kurzreferate

B.C.E., 5, 1, 12. U.D.C. No. 66.045:1-974

TIEFTEMPERATURWÄRMEAUSTAUSCHER

von P. M. Shuftan

Eine Übersicht der verschiedenen in Tieftemperaturgasabscheidern benutzten Arten von Wärmeübertragungsgeräten, ihrer Betriebsverhältnisse und der in ihrer Planung verfolgten Ziele. Der Einfluss der Verunreinigung, die Bedeutung von niedrigen Wärmegradunterschieden an der Warmseite (Drucklufteintritts- und Stickstoffaustrittspunkt), niedrige Druckverluste und kleine Wärmegradunterschiede in Aufwärmern sind Gesichtspunkte von besonderer Bedeutung fur die Wirtschaftlichkeit von Tieftemperaturvorgängen.

B.C.E., 5, 1, 15. U.D.C. No. 662. 995-974: 66.045. 1-974

DIE ANWENDUNG VON REGENERATOREN UND REVERSIERWÄRMEAUSTAUSCHERN

von D. E. Ward

Tieftemperaturwärmeaustauscher müssen selbstreinigend, kompakt und billig sein und die Fähigkeit besitzen, mit sehr kleinen Wärmegradunterschieden zu funktionieren. Die relativen Vorteile von Reversieraustauschern und Regeneratoren für Zwecke der Tieftemperaturwärmeübergabe werden verglichen.

B.C.E., 5, 1, 18. U.D.C. No. 66.045.1:66.074.8

DIE ANWENDUNG UND DIE AUSFÜHRUNG VON WÄRMEAUSTAUSCHERN MIT RIPPENVERBUNDENEN PLATTEN

von W. H. Denton und D. E. Ward

Die Vorteile des Wärmeaustauschers mit rippenverbundenen Platten für die Wärmeubergabe und Gasreingung in Tieftemperaturgasabscheidern werden besprochen. An Hand von Beispielen werden Verfahren zur Ausführung dieser Aggregate fur spezifische Zwecke der Wärmeübertragung und Druckfällung, sowie zur Schätzung der Wirkungen einer eventuellen Strömungsmisverteilung angegeben.

B.C.E., 5, 1. 27. U.D.C. No. 66.045.1

WÄRMEAUSTAUSCHER MIT SEKUNDÄROBERFLÄCHEN

von A. G. Lenfestey

Die Bauweise von Wärmeaustauschern mit Wellplatten und die möglichen Anordnungen von Oberflächen dieser Art, sowie auch ihre Anwendungen in Tieftemperaturvorgängen und ihre Wärmeübertragungseigenschaften werden beschrieben.

B.C.E., 5, 1. 33. U.D.C. No. 662.995

EINE VEREINFACHTE REGENERATORTHEORIE

von W. F. Schalkwijk

Eine einfache Gleichung zur Berechnung der Leistung eines Regenerators wird abgeleitet. Die Gleichung beruht darauf, dass die meisten Regeneratoren in der Praxis ein lineares Temperaturfeld in ihrer Mitte aufweisen.

B.C.E., 5, 1, 35. U.D.C. No. 662.995:66.012.7

ALLGEMEINE BETRACHTUNGEN ÜBER DIE LEISTUNG UND KOSTEN VON REGENERATOREN

von H. London

Der Einfluss verschiedener Faktoren auf die Ausführung von Regeneratoren, wie z.B. Druckgefälle, Strömungsverteilung, längsweise Wärmeleitung und Gasaufhaltungsvermogen, wird besprochen. Eine Reihe von Gleichungen wird abgeleitet ; diese bieten einen Weg zur Planung von Regeneratoren zu Mindestgesamtkosten.

B.C.E., 5, 1. 37. U.D.C. No. 66.045.2:66.048.2

DIE DER KONDENSATION VON GEMISCHTEN DÄMPFEN BEIFOLGENDE WÄRMEÜBERGABE

von G. G. Haselden und W. A. Platt

Eine Kritik der verfügbaren Verfahren zur Berechnung der Kondensationsschichtkennziffern fur geringe Wärmegradunterschiede. Die notwendigen Konstruktionsmerkmale eines für Tieftemperaturgefälle geeigneten Kondensators werden angegeben und eine wirksame Ausführung wird beschrieben.

B.C.E., 5, 40. U.D.C. No. 66.045.1:66.074.8: 66.012.7

ALLGEMEINE WIRTSCHAFTLICHKEIT VON WÄRMEAUSTAUSCHERN MIT RIPPENVERBUNDENEN PLATTEN

von W. H. Denton

Die Energie- und Kapitalanlagekosten von Wärmeaustauschern stellen einen bedeutenden Teil der Gesamtkosten von unter etwa 100° K verlaufenden Verfahren dar. Beispiele davon sind Luft- und Wasserstoffdestillieranlagen. An Hand von Beispielen der Optimalausführung solcher Austauscher wird gezeigt, dass die Kosten der Kühlenergie und der Druckverluste und die Kapitalkosten stets in einem Verhältnis von 3 : 1 : 2 zueinander stehen.

Резюме

B.C.E., 5, 1, 12. U.D.C. No. 66.045:1-974

НИЗКОТЕМПЕРАТУРНЫЕ ТЕПЛООБМЕННИКИ

П. М. Шуфтан

Обзор различных типов установок теплопередачи, применяемых в низкотемпературных газоотделителях, режима их работы и принципа их конструкции. Влияние загрязнения, небольшие температурные различия в «теплом конце», т.е. при входе сжатого воздуха и выходе азота, низкие потери давления и небольшие температурные перепады в подогревателях являются особенно важными факторами в экономике низкоетмпературных процессов.

B.C.E., 5, 1, 15. U.D.C. No. 662. 995-974: 66.045. 1-974

ПРИМЕНЕНИЕ РЕГЕНЕРАТОРОВ И РЕВЕРСИВНЫХ ТЕПЛООБМЕННИКОВ

Д. Э. Уорд

Низкотемпературные теплообменники должны быть самоочищающимися, компактными, недорогими и способными работать при весьма небольших температурных различиях. Сопоставлены относительные достоинства реверсивных теплообменников и регенераторов для целей низкотемпературной теплопередачи.

B.C.E., 5, 1, 18. U.D.C. No. 66.045.1:66.074.8

ПРИМЕНЕНИЯ И КОНСТРУКЦИЯ ПЛАСТИНЧАТО-РЕБРИСТЫХ ТЕПЛООБМЕННИКОВ

У. Х. Дентон и Д. Э. Уорд

Обсуждаются преимущества пластинчато-ребристых теплообменников для теплопередачи в низкотемпературных газоотделителях. Приводятся способы и примеры конструкции таких агрегатов для определенных задач относительно теплопередачи и перепада давления, а также оценки влияния нерационального распределения потока.

B.C.E., 5, 1. 27. U.D.C. No. 66.045.1

ТЕПЛООБМЕННИКИ С ВТОРИЧНЫМИ ПОВЕРХНОСТЯМИ

А. Дж. Ленфестей

Автор описывает конструкцию теплообменников с волнистыми пластинами и некоторые возможные распределения поверхностей такого типа, их применение для низкотемпературных процессов, а также их теплопередачную характеристику.

B.C.E., 5, 1, 33. U.D.C. No. 662.995

УПРОЩЕННАЯ ТЕОРИЯ РЕГЕНЕРАТОРОВ

В. Ф. Шальквайк

Выведено простое уравнение для расчета к.п.д. регенератора, основанное на том, что большинство регенераторов на практике имеют линейное температурное поле в своем центре.

B.C.E., 5, 1, 35. U.D.C. No. 662.995:66.012.7

ОБЩИЕ СООБРАЖЕНИЯ О К.П.Д. И СТОИМОСТИ РЕГЕНЕРАТОРОВ

Х. Лондон

Рассматривается значение некоторых факторов, влияющих на конструкцию регенераторов, как напр. перепада давления, распределения потока, продольной теплопроводности и газоудерживающей способности. Приводится ряд уравнений, обеспечивающих способ конструкции регенераторов минимальной общей стоимости.

B.C.E., 5, 1. 37. U.D.C. No. 66.045.2:66.048.2

ТЕПЛОПЕРЕДАЧА, СОПРОВОЖДАЮЩАЯ КОНДЕНСАЦИЮ СМЕШАННЫХ ПАРОВ

Дж. Дж. Хэзелден и У. А. Плэтт

Критика существующих способов расчета коэффициента конденсационных пленок для небольших температурных различий. Дается краткий обзор необходимых конструктивных особенностей конденсатора для низкотепловых напоров и описание эффективной конструкции такого конденсатора.

B.C.E., 5, 40. U.D.C. No. 66.045.1:66.074.8: 66.012.7

ОБЩАЯ ЭКОНОМИКА ПЛАСТИНЧАТО-РЕБРИСТЫХ ТЕПЛООБМЕННИКОВ

У. Х. Дентон

Стоимость электроэнергии и капиталоемкость теплообменников представляют значительную часть общей стоимости процессов, протекающих при температурах ниже около 100° К. Примерами этого являются воздухо — и водородоперегонные установки. На примерах оптимальной конструкции таких теплообменников показано, что стоимости холодильной энергии, потерей напора и капиталовложения всегда находятся в отношении 3 : 1 2 соответственно.

DIGEST OF INDUSTRIAL AND TECHNICAL DEVELOPMENTS

Rising Soviet Potential for Automatic Control

THERE is little evidence of current work in the Soviet Union on the more advanced forms of control of chemical processes but a great potential is being built up for further developments in the form of large well-equipped research organisations and considerable numbers of engineers trained in these fields. Earlier this year a team of six British engineers visited five Soviet research institutes concerned with control systems. Of these two had been established for only two years and one for three. At the Central Institute of Automation of the Ukraine, at Kiev, which then had 700 employees, there were plans for new buildings to be completed in three to four years providing for an increase in the number of personnel to 4000. This Institute —like the central one in Moscow—will study developments of automation in heavy industry, including chemical plants.

In their report,* the team issue a warning against complacency based on present-day comparisons. Institutes only two or three years old cannot yet have made their weight felt in industrial applications so that present achievement is no proper measure of the large potential for progress. The groups in Britain for research and development in civil applications of automatic control are very much smaller than those now existing in the Soviet Union, and the contrast is still greater in respect of expectations for the near future. They sombrely conclude that in a few years the application of automatic control in Soviet industry will have overtaken that in this country. The stage will then be reached at which leadership will depend upon the breaking of new ground. However, it should be noted that there will be an excellent opportunity to develop communications with Russia by attending the conference of the International Federation of Automatic Control in Moscow, in July 1960.

One member of the team, PROFESSOR TUSTIN, also visited Poland. Here it appeared that the work, though well chosen, was at present more preparatory than effective in actual industrial application.

* "Automatic Control in Soviet Industry." Department of Scientific and Industrial Research.

Piping New

CHEAPEST form of transport for fluids overland is usually by pipeline. However, the main obstacle is the very high capital cost of construction. It is therefore of considerable interest that *Pipes and Pipelines* have initiated a new section devoted to pipeline engineering (October, 1959). This opens with a review by I. ALEXANDER of the trends to thinner pipes, larger ones, "tailor-made" pipes and the use of welded construction. Transport costs for the pipes themselves are so heavy that there is strong incentive to find means of reducing them. On the other hand pipe-making plant particularly for large diameter welded linepipes and for large seamless pipe is so huge that it cannot be relocated near the point of construction of a line. Processes are however being devised using machinery which is reasonably transportable and which could be located at a port or railhead near the point of construction. One remarkable example of this type of machinery is the Spiral or Helical welding type of machine which uses steel strip in large coils. By deforming the strip through certain prescribed angles, it can produce pipe without the use of a great deal of electric or other energy and without much limitation on lengths produced. With present methods, however, this type of machine is capable of producing good linepipe at the low conversion cost of between $6 and $8 a ton, including all costs (based on U.S. wage levels) other than the cost of the parent metal itself.

In other contexts, uses of piping are also likely to be extended by the improvements in quality and application of PTFE lining—claimed to be resistant to temperatures up to 260°C, unaffected by almost all chemicals and almost universally useful for corrosion-proof service. The only reagents to which it is vulnerable are molten alkali metals, and free fluorine at elevated temperatures and pressures. By new processing techniques somewhat resembling those used with ceramics it has been applied to piping ranging from 1 in. to 6 in. diameter.

Growing Attention to Standards

AS the frontiers of science and technology advance, the variety and complication of plant increases. It is understandable that men in many countries should strive to limit the diversity in design which arises from differences in convention rather than in fundamentals and also to ensure that units mean exactly the same thing to all workers. Evidence of this trend to standardising reaches us continuously. In *B.S.I. News* for November, 1959, MR. W. R. BESWICK, chairman of the B.S.I. Chemical Engineering Industry Standards Committee contributes a review of the brochure on relevant specifications which we noted last month. In this MR. BESWICK refers to the great importance attaching to structural safety of vessels under varying degrees of pressure, vacuum or temperature; to materials of construction which will withstand surface attack; and interchangeability of valves, flanges and fittings in order to relieve the already onerous task of the works maintenance engineer.

It is noteworthy that a recent copy of *Chemische Technik* (Vol. 11, No. 10, 1959) reveals the considerable efforts now being made in East Germany to tidy up variations in views on both plant design and the meanings of technical terminology. Two articles are devoted to standardising equipment and procedure in the chemical industry while a further one proposes definitions for meanings of words used in technical terminology. The latter is considered necessary because there are varying interpretations in the literature. In the U.S.A. the National Bureau of Standards is expanding its services by now distributing 60,000 samples of standard materials a year to other laboratories for use in controlling chemical processes and in maintaining the accuracy of apparatus and equipment. A total of over

600 different materials are available from the Bureau—principally chemicals, ceramics, metals, ores, and radioactive nuclides. They are intended to promote uniform measurements of heat and temperature, define the colours of paints, and calibrate instruments that control the composition of metals and motor fuels.

In the high-pressure programme, effort is being directed towards extending to higher pressures those techniques that are now used to generate and measure pressures below 200,000 psi. These pressures can now be measured by dead-weight-loaded pistons that are not packed to prevent pressure fluid from leaking. Instead, the clearance between piston and cylinder is made so small that leakage is kept within a few cubic inches per month. This type of apparatus may be found usable, with a fluid pressure medium at pressures as high as 350,000 or 400,000 psi.

Improving the Filtrability of Sugar Solutions

IT appears likely that many mills could improve the filtrability of their raw sugars by reducing the amount of suspended matter passing over from the clarifiers in the clarified juice. After extensive chemical and chemical engineering investigations, C. W. DAVIS, Chief Chemist of the Australian Colonial Sugar Refining Co. came to this conclusion and proposed methods for acting on it. In papers published by the *International Sugar Journal** MR. DAVIS proposes that if laboratory clarification tests indicate phosphate deficiency in juice, 20-40 p.p.m. of phosphorus should be added as superphosphate, since the ideal content of available phosphate in juice is thought to be around 150 p.p.m. (as phosphorus). In juices containing considerably less than this figure it does not appear economical to add more than 40 p.p.m. The juice should be heated and degasified. The lime as saccharate should be added into a positive mixing device, which gives sufficient mixing to flow a representative sample past the pH electrodes, but does not impart sufficient turbulence to break the floc. It is essential to control pH automatically, and ensure that available phosphate in clarified juice is below 15 p.p.m. (as phosphorus). Some aspects of this control are referred to below but it should also be noted that it is advisable to protect the electrodes against excessive temperature by passing some of the boiling juice through a small cooler to the electrode cell.

However there is no substitute for analysing an existing process carefully and checking crucial points such as the adequacy of phosphate content in the mixed juice, whether lime is being properly mixed with the juice or the flocs being damaged due to turbulence. Finally, backing this work, must be fundamental research into the natures and origins of the substances causing raw sugar solutions to filter badly.

* Oct. 1959, page 300; Nov. 1959, page 332.

Differential pH control in Sugar Production

THE difference in value between a preset and a measured pH may be used—alone or with other information—to control sugar production at several stages. In the pre-mixing the signals can correct the quantity of added lime. At the first saturation stage, the same method may regulate the input of CO_2. It is used alone, however, only if the control needed is limited to rather narrow limits, for example, a range of ±1.5 pH. Otherwise, according to K. CIHAL* it is usually coupled with coarse manual control by an operator. If fully automatic control is required a second pH regulator has to be provided and set to react to large deviations of the pH value, such as ±10 pH units. Another method derives the coarse control (70-80%) of CO_2 from the flow of the stirred juice and the fine control (20-30%) from the differential pH readings. In a Polish plant using this technique, pH of the first saturation juice was thus maintained between 10.8 and 11.2 pH units.

The differential pH meter has also successfully been used for automatic control of the input of CO_2 in the second saturation process of the juice. The reference solution of the meter is set with respect to the minimum number of calcium ions in the juice and the difference of pH readings is used to control the setting of the CO_2 valve. To ensure that the minimum level of Ca in the product is continuously maintained the setting of the valve is changed, in response to the change of the pH valve of the juice produced, only if this occurs simultaneously with the increase of the number of the Ca ions in the product. While the percentage of Ca is decreasing no resetting of the CO_2 valve will take place. The quantity of Ca in the product is continuously monitored by a flame photometer.

* *Listy Cukrovarnicke*, No. 10, 1959, p. 230.

Clear Away All Explosive Dust

IMPROVED "housekeeping" in factories is demanded by a new regulation which comes into force on February 1 1960. Accumulations of explosive dust must now be cleared away wherever they occur—in corridors, stairways and the like, not simply in workrooms. This is the effect of Section 7 of the Factories Act, 1959 which amends the principal Act. The latter had required certain precautions only where dust liable to explode on ignition might escape into the workroom. The new section extends this statutory provision to *any* accumulation of such dusts.

Fluidised Bed Nuclear Reactor

IS it feasible to design a reactor for nuclear power production using a fluidised bed of fuel pellets ? The project has been examined for the U.S. Atomic Energy Commission by the Martin Nuclear Division on the basis of having the bed circulate in a confined core region. The fluid would act as a fluidising, coolant and reactor moderator medium. Previous studies of such a system had suggested several advantages. The fuel burnup would be uniform, the conversion ratio higher, while the fuel element could be an unclad uranium dioxide pellet of simple shape. With continuous processing, variation of flow and of fuel enrichment, reactivity is controlled without the use of control rods. In the first phase of the examination there has been major emphasis on the uranium-plutonium dioxide, light water systems and preliminary studies on the organic moderated system. In addition control problems were examined by analogue simulation to arrive at criteria for optimising conditions. Erosion tests in water showed very little fuel weight loss. Corrosion tests indicated that the unclad pellets would be stable in water of high temperature reactor grade having concentrations of oxygen up to one cubic centimetre/litre. There are however some difficulties due to abrasion of unclad fuel pellets and the effects of these abrasion fines in the primary loop. Nevertheless, the conclusion was that the concept offered considerable promise and a report on the work* makes detailed suggestions for the further investigations necessary to provide the design data for a reactor system of this type.

* No. MND-FBR-1696, UC 81 available from the Department of Commerce, Washington 25, D.C. $3.50

BCE 6698 for further information

"Home-made" liquid air for research and industry

PHILIPS GAS REFRIGERATING MACHINE

Produces 5 litres (1.1 gallon) of liquid air per hour at surprisingly low cost; power consumption of the motor drive is only 6 kW.

Full output within 15 minutes from start up; continuous supervision unnecessary - the "fail to safety" principle is applied throughout.

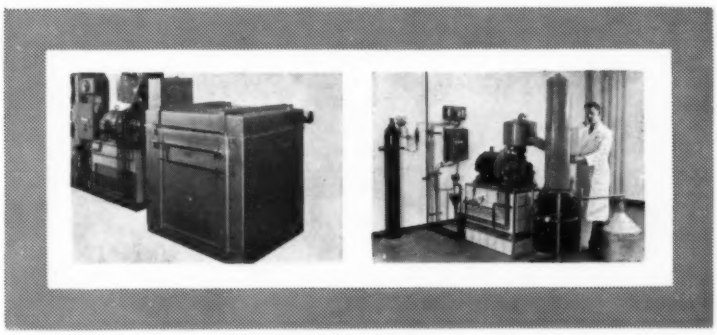
No compressors or pumps for sucking in the gas or tapping off the liquefied product; air liquefies on stationary cold surface; liquid air drains off by gravity.

Moisture and carbon dioxide removed by freezing; no chemicals, no reactivation required.

ACCESSORIES FOR PHILIPS GAS REFRIGERATING MACHINE

Air separation column for production of 4—5 litres (1 gallon) of liquid nitrogen per hour (more than 500 litres (110 gallons) per week).

380 litres cold box for storage, testing or cold treatment of materials from room temperature down to -160 °C (-255 °F); refrigerating capacity at -100 °C: 1110 kcal/hr (4440 BTU/hr) and at -160 °C: 640 kcal/hr (2560 BTU/hr) respectively.



Sales and service organizations all over the world.

PHILIPS

Full information about this simple, reliable installation supplied on request by:
N.V. PHILIPS' GLOEILAMPENFABRIEKEN - Scientific Equipment Division - EINDHOVEN - HOLLAND

Sole distributors in the U.K.:
Research and Control Instruments Ltd., 207 King's Cross Road - LONDON W.C. 1

Which Shales Gum Up Coal Preparation?

DURING coal cleaning by wet processes, some shales readily break down in the washeries. This may lead to difficulties in clarifying the washery water, filtering the froth flotation tailings, or even in conveying the discards. Accumulation of the mud on rollers and driving drum can bring the belt to a halt. At the National Coal Board, a Working Panel was set up to investigate the breakdown in water of the discards from coal preparation plants, as well as the flocculation and filtration of the resultant slimes. A first report of this work has now been published by I. BERKOVITCH, M. MANACKERMAN and N. M. POTTER in *Journal of The Institute of Fuel*, Vol. XXXII, No. 227, 1959, page 579.

To assess the liability to breakdown, a standardised empirical test was developed. Results from this test compared with those from a pilot-scale jig washer were found to give a fair indication of the degree of breakdown of the shales as they passed through the plant; they would probably provide a relative measure of disintegration for standard conditions in a given comparable plant. This work is being extended to production units. An interesting discovery during this programme was that there are wide variations in the times for which particles are retained in a washer. Associated experiments in a Greaves box jig washer used pieces of coal and shale which had been painted to aid identification. Most pieces were retained for 2 to 5 min. with an average time of about 4 min.; individual pieces were retained for times ranging from 1½ to 12 min. By observing the behaviour of the marked pieces as they entered the washer it was obvious that different pieces were taking different paths through the plant; some disappeared immediately into the bed whilst others remained in view for some time.

Flameless Brazing by Hot Gas

A STREAM of hot inert gas—mainly argon—is the effective heating agent in a new brazing torch now being patented by the U.K. Atomic Energy Authority. Its development arose from the difficulties found in attempts to join reactive metals such as uranium and thorium to other metals by means of oxy-gas brazing. The reactive metals are easily contaminated by the oxygen, nitrogen, carbon monoxide, carbon dioxide and water vapour produced by an oxy-gas torch. This led to the concept of using an argon-rich gas raised to (unspecified) high temperatures by an intense electric arc. A supply duct passes the gas through the arc which is initiated by a high frequency discharge and maintained by a 60 volt d.c. supply, one electrode being kept at earth potential. A fine pointed jet is produced which blankets the surface under treatment and protects it from contaminating gases. Temperature is controlled by adjusting the power input. The torch can easily be adapted for duties such as local heat treatment and stress relieving or softening metals for bending. It should be suitable for use when fabricating in zirconium, titanium and beryllium, or for accurate cutting of stainless steel at high speeds.

Disentangling the Processing of Wool

THE wool textile industry is not normally considered as a field for applying chemical engineering methods but we recently received news of interesting developments in Australia and Britain. From Melbourne, DR. F. G. LENNOX* has informed us of his plans to increase knowledge on the structure and chemistry of wool and to devise new wool processes arising out of this work. A three-year fellowship has been offered for a chemical engineer to carry through both this development and the "application of general chemical engineering principles in the wool textile industry."

Meanwhile at the Wool Industries Research Association in Leeds, there has been great progress in design of new types of driers, particularly for loose wool, since the end of the war. Compared with the performance of pre-war equipment the reduction in steam consumption is from 4-1.5 lb. of steam per lb. of water evaporated. In addition the new machines give much less entanglement of the wool and cheaper combing. DR. A. B. D. CASSIE, Director of the Association, points out that wet treatment of masses of fibres at any stage is essentially a chemical engineering problem. There is, for example, the difficulty of transferring heat through masses with such good heat insulation and retaining constant concentration of chemical components such as dyestuffs in a liquor flowing through what is essentially an absorption filter. Again, improving the scouring and milling of pieces of cloth involves studies of the expression of liquors by nips and the danger of cracking soaps by changing too rapidly from a solution of one strength to one of much lower strength.

The Association has also developed a metering pump to deliver controlled quantities of oil independent of viscosity or state of emulsification of the oil.

* Chief of the Division of Protein Chemistry, Wool Research Laboratories, Commonwealth Scientific and Industrial Research Organisation.

Chemical Exports Rise

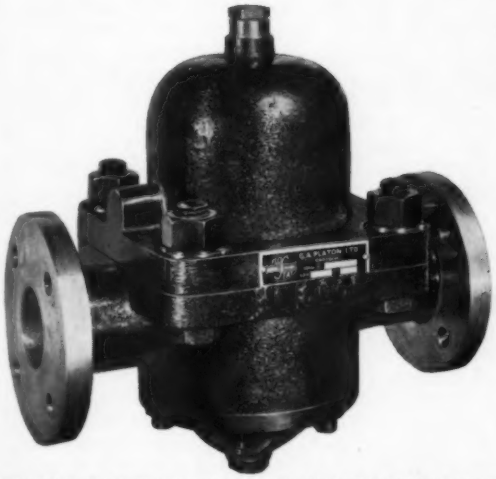
IN the first 10 months of 1959 chemical exports were 11.1% higher than in the corresponding period of last year. This must be set against the fact that last year there had been a fall of 2%. Nevertheless, the rise in exports of chemicals was the largest among the main categories of manufactures. Outstanding gains were shown by plastic materials—about one eighth of the total—which registered 27.0% and the composite group chemical elements and compounds—a quarter of the total—at 15.6%.

Sintering Aluminium/Aluminium Oxide Powder

SINTERED masses cannot be produced from aluminium powder containing oxide by ordinary sintering treatments, that is by heating to temperatures below the melting point and holding at these temperatures; it is found essential to submit the sintered mass to a subsequent hot-forming process. Why is this? At a recent symposium on powder metallurgy* E. M. MODL-ONITSCH suggested that this was connected with an unexpected temperature rise found on heating the cold-pressed powder in nitrogen to 525°C. The ground aluminium oxide $Al_xO_y(OH)_z$ decomposes upon being heated—apparently very abruptly in the temperature range 525°-550°C.—forming $Al_xO_{y+n}(OH)_{z-2}+nH_2O$. The metallic aluminium then reacts with this water giving $Al_2O_3+3H_2$. The last reaction is markedly exothermic, which would account for the sudden temperature increase. Probably in the sintering adhesion forces are effective in the first instance, with atomic and molecular regroupings taking place in addition and doubtless assisting the process. The temperature increase of 100°C. might explain the fact that it is possible at all to sinter a system consisting of two constituents with such widely differing melting points as aluminium oxide and pure aluminium, into a more or less solid mass at the relatively low sintering temperature of 550°-600°C.

FLOW CONTROL FOR 1960

THE MOST SIMPLE AND INEXPENSIVE METHOD OF FLOW CONTROL
USES PART OF THE PRESSURE ENERGY OF THE FLUID
TO OPERATE THE CONTROL VALVE ENTIRELY AUTOMATICALLY



STANDARD FLOSTAT (1½" size at £48) BALANCED DOUBLE-BEAT POPPET VALVE AND METAL BELLOWS. FACTORY CALIBRATED TO USER'S SPECIFICATION WITH NORMAL ACCURACY 1%. FOR CONTROL OF:—

SERVICE FLUID DISTRIBUTION (Water, Air, Brine, Solvent, Steam, etc.) BLENDING AND DILUTION PROCESSES FEED TO FILTERS, EXTRACTORS, etc.

The controller employing this principle is the Flostat, first made in 1955 but now of proven value in many industries—from Food to Atomic Power.

The Flostat self-acting flow controller applies the pressure differential at a built-in metering orifice to a pressure-responsive element which is directly connected to a balanced control valve. Variations in upstream and downstream pressure tending to alter the flow rate through the orifice result in very rapid readjustments of the valve opening so that the flow rate is kept constant.

Standard Flostats with factory calibrated (non-adjustable) setting have been found to be ideal for many fixed flow control applications where the high cost of power-operated controllers could not be justified. It became clearly apparent that the modest cost of Flostats could soon be recovered by improving plant efficiency (e.g. filters), by eliminating manual adjustments (e.g. constant blending) or by preventing waste of service fluids.

The point has now been reached when 100 Standard Flostats have been installed on a single new plant—an impressive testimonial to the robust reliability of an essentially simple design.

A natural development from the Standard Flostat has been the Type 'V' (variable orifice) Flostat. This is fitted with a calibrated control which enables the rate-of-flow setting to be altered within a 10:1 flow range. Thus the advantages of the Flostat method of flow control are now made available to many more industrial processes.

The need for an even more simple rate-of-flow limiting device, as a means of improving plant safety, has led to the development and production of the Type 'B' Flostat. In place of the bellows and poppet valve of the Standard instrument, the Type 'B' Flostat uses a "Bellofram" and spring with a butterfly valve which, in normal flow conditions, offers the minimum resistance to flow through the instrument.

FLOSTAT TYPE 'V' (1" size at £64) STANDARD FLOSTAT WITH VARIABLE ORIFICE FITTED. CALIBRATED AS REQUIRED—MAXIMUM] FLOW RANGE 10:1 FOR 1% ACCURACY. FOR CONTROL OF:—

BURNERS
CHEMICAL PLANT UNITS (Stills, Evaporators, etc.)
CHEMICAL PROCESSES OF ALL KINDS.

FLOSTAT TYPE 'B' (4" size at £56) BUTTERFLY VALVE WITH "BELLOFRAM" AND SPRING. FACTORY CALIBRATED GOVERNOR IMPOSES MINIMUM RESTRICTION TO NORMAL FLOW BUT PREVENTS EXCESSIVE FLOW AS PRESSURE INCREASES. FOR LIMITATION OF FLOW:—

TO PROTECT PLANT AND METERING EQUIPMENT FROM SURGES
TO PREVENT BUILD UP OF STATIC CHARGE

Also available are: The Flostat Minor, for very small gas and liquid flows; Gapmeter variable-area flowmeters; Flo-scan flow operated alarm switches; a comprehensive consultation and design service for new Flow Measurement and Control installations.

The full range of G. A. Platon instruments will be demonstrated at the **1960 Instruments, Electronics and Automation Exhibition** being held at **Olympia, 23rd to 28th May** (inclusive) from 10 a.m. to 6 p.m. **Flow Control for 1960—G. A. PLATON LTD.—STAND G 317.**

G. A. PLATON LTD.
323a WHITEHORSE ROAD
CROYDON, SURREY
THOrnton Heath 1631

More generally, DR. MODL-ONITSCH referred to the fact that extensive work carried out in recent years, especially with sinter models, has shown that adhesion forces, activation energy, diffusion processes, and plastic flow are the principal factors controlling the sintering process. Which factors and how many of them are operative in a particular sintering system then depends on the number and nature of the constituents to be sintered.

* Organised by the Powder Metallurgy Joint Group of the Iron and Steel Institute and the Institute of Metals.

Weighing in Theory and Practice

AUTOMATIC batch weighing and economics are closely related. Suppose that a factory is producing fertilisers and delivering the product in bags, which are weighed-off with automatic batch weighers. If underweight is not permitted the machines must be set to deliver a certain average overweight. This cannot be avoided, even if the weighing machines are operated with the greatest possible care. It is also known that the weights delivered are usually more accurate when the speed of weighing is lower. But the lower the speed, the more weighers are needed to obtain the required number of bags per unit time. What will be the economical optimum ? Alternatively, how many machines should be installed to give the best result economically ? The answer to this question depends in the first place on the margin between the cost of production and the sales price. In the second place, it depends on the relation which exists between the speed and the accuracy of weighing for a given batch weigher.

DR. A. WALLER in a recent paper to the Fertiliser Society set out these considerations and went on to derive these relationships on theoretical grounds for the main types of automatic batch weighers which have been developed so far. At the same meeting W. H. BONSER presented a general review of automatic weighing equipment for the fertiliser industry while R.I.O. Passmore reported careful tests on the performance of equipment. An important conclusion was that weighing equipment used for trade purposes, and therefore subject to current inspection and control, conforms to the limits of error required by the Weights and Measures Regulations provided due attention is paid to the condition and operation of the machines. Process weighing machinery, on the other hand, does not present such a happy picture. The standard of weighing is not as high as in trade weighing machines and it would appear that the weighers are not built to achieve the same order of accuracy. There seem to be several morals in this.

Removing Carbon Monoxide from Fuel Gases

"MOST of the questions people have been asking about the process are answered in this paper." Who could want a more generous tribute than this comment in discussion on a research report* devoted to the water gas shift reaction ? The reaction has of course been used for many years in the chemical industry but dealing with clean gases. In applying it to coal gas, or more particularly to carburetted water gas the main problem is the fouling of the catalyst. This is attributed to the interaction of conjugated diolefins, nitric oxide and oxygen on the surface of the catalyst. Fouling can, however, be minimised by removing these constituents by a pretreatment catalyst. G. S. CRIBB and J. D. F. MARSH—in the paper GC66 which received the plaudits—reported that they had examined several such catalysts. The most promising proved to be nickel subsulphide, and life tests are in progress of shift catalysts with and without prior protective catalysts.

If the carbon dioxide formed in the water gas shift reaction is removed from the treated gas in order to maintain its calorific value, the specific gravity of the product becomes considerably less than that of the original gas. For this reason, and because of the high cost of carbon dioxide removal, close attention was paid to the extent to which this is desirable under a variety of conditions. The extent to which carbon dioxide need be removed can be reduced by using light distillate as a feedstock and by the admixture of rich gas to increase the calorific value. For partial removal, the most promising processes are those based on ammonia or activated potassium carbonate as the absorbent. Properties of the reagents such as the absorption rate for carbon dioxide, the resulting vapour pressures and the heat requirements for regeneration are being studied at The Gas Council's London Research Station using a pilot plant.

* Autumn Research Meeting, Institution of Gas Engineers.

An Approach to "Research Economics"

THOUGH it is necessarily couched in very general terms, the attempt by CARTER and WILLIAMS* to set out a rational procedure for taking research and development decisions should prove valuable to many firms. The authors advocate that decisions about research and development should be made, not only with regard to the productive purposes of the firm, but also with a clear appreciation of what resources the firm can command in technology, management, and finance. Furthermore, research projects should be chosen by men or committees with an appreciation of the production, selling, and financial problems of the firm, as well as of the scientific possibilities. Rough estimates should be made of the *yields* from projects considered both capable of scientific solution and of commercial significance. This can give priority lists of projects at each of the stages of research and development, and be the basis of the research programme. PROFESSORS CARTER and WILLIAMS are well aware that even the most efficient firm cannot attain precision in these assessments and the fixing of a research budget often appears to be little more than guesswork. Nevertheless they think it important that managers should do what they can to make estimates of the yield from the various uses of the firm's funds, including research and development.

Apart from the immediate and direct economic value, this approach helps in maintaining staff morale when a project has to be abandoned. The workers concerned should participate in this decision. This is possible only if they can be persuaded to look at the full range of relevant facts, commercial as well as technical; that is, if they learn to think in terms of "research economics."

* Science in Industry. Policy for Progress. Third Report of the Science and Industry Committee, by C. F. Carter and B. R. Williams, 21s. net. Oxford University Press, 1959.

Vacuum Abhors a Leak

A LEAK may be more than a nuisance casting extra load on pumps. Inflow of air or moisture may cause decomposition; outflow from a pressure plant may form a risk to health. This is notably true of nuclear energy equipment. Methods successfully used for final checks on their pressure or vacuum-tightness have been reviewed* by W. STECKELMACHER. The testing may be carried out by positive pressure or by vacuum methods. In either case it is liable to take a great deal of time and the basis of the

Announcing

CAMPBELL CHEMICAL ENGINEERING CO. LTD

as from January 1st 1960
have changed their name to

Process Equipment Ltd

This has been found necessary to avoid confusion with other firms of a similar name

Process Equipment Ltd

DESIGNERS AND MANUFACTURERS OF FLUID MIXING EQUIPMENT AND SPECIALISED PLANT FOR THE CHEMICAL INDUSTRY

SHEPLEY WORKS, AUDENSHAW, NR. MANCHESTER Telephone: DENTON 3831/5
LONDON OFFICE: 17 STRATTON ST., W.1. Telephone: GROSVENOR 3722

Coal Chemicals: Ideas Invited

THE Wilson Committee which was set up to review recent work and make recommendations on coal-based chemical processes is inviting ideas. In reply to questions in the House of Commons, the Minister of Power, MR. RICHARD WOOD, stated that it would welcome further written memoranda from people interested in research and development on processes making chemicals, gases and oil from coal and in the organisation of such work. Material should be addressed to the Secretary of the committee, MR. W. R. G. BELL at the Ministry.

results must be closely specified including the pressure differentials employed in a positive pressure method. The same hole, tested by the two methods, will give two different leakage rates. Assuming viscous laminar flow, the two rates are related by the expression $L_p = (P^2 - I)\ L_0$ where L_p is the leakage rate at positive pressure P (in atmospheres) and L_0 is the vacuum leakage rate.

For vacuum the most sensitive of all the leak-detection instruments is the mass spectrometer. This instrument does not depend for its operation on pressure differentials but separates, and indicates the mass of, the ions formed by the test gas—usually helium—and the ions formed by the residual gases passed in from the vacuum system. The instrument is capable of detecting leakage rates of less than 10^{-6} lusecs†. Next to it in sensitivity is the palladium barrier leak detector, employing hydrogen as the search gas and a hot ionisation gauge as the sensing element. In the oxygen leak detector the operating principle is based on the fact that the presence of oxygen in a diode causes a decay of the electron emission from the cathode. Using a Pirani gauge as the sensing element are the differential method, the hydrogen-Pirani method, and the charcoal-Pirani method. In addition to these and other vacuum test methods, there is, however, also interest in pressure methods often using simpler equipment, which MR. STECKELMACHER has also described with indications of their sensitivities.

* *Nuclear Engineering* 1959, Vol. 4, No. 43, p. 450.

† A leak allowing a gas mass flow rate which causes a pressure rise of 1 micron Hg per second in a volume of 1 litre = 1 lusec. 1 c.c./min (n.t.p.) = 12.67 lusecs.

Creep in Thick-Walled Cylinders

THERE is sufficient common ground between the fields covered by chemical and mechanical engineers to make it always interesting to look through the papers presented to such bodies as the American Society of Mechanical Engineers. At their Annual Meeting in December, 1959, IAIN FINNIE examined the methods available for predicting steady-state creep stresses and strain rates in a thick-walled tube under combined internal pressure and axial load. Although an analytical solution of the equations, in closed form, is not possible, MR. FINNIE has tabulated numerical solutions to the equations for a range of values of the variables. In addition the author gives a trial method of calculation using the tabulated solutions as the starting point. It is important to note that the stress distribution in a thick walled cylinder under this combined load cannot, in general, be obtained by attempting to superimpose the stresses due to the separate loads. In his paper (No. 59—A-57) the equations required for a stress analysis have been developed by the author and then solved with a digital computer for a number of tubes, creep laws, and loading conditions.

DIARY

Institution of Chemical Engineers

January 5. The Geological Society. At Burlington House, London, W.1. "Heat Transfer to Two-phase Gas/Liquid Systems—Part 1," by J. A. R. Bennett, J. G. Collier, H. R. C. Pratt and J. D. Thornton. "Pressure Drop and Hold-up in Two-phase Flow," by J. G. Collier, J. D. Thornton and J. A. R. Bennett. **5.30 p.m.**

January 14. *Scunthorpe Group.* At John Leggott's Grammar School, Scunthorpe. "Automatic Analytical Instruments." **7.30 p.m.**

January 20. *North Western Branch.* At The Houldsworth School of Applied Science, The University of Leeds, Leeds 2. "Refrigeration Practice," by Mr. Perry. **6.30 p.m.**

January 26. *North Western Branch.* At The Manchester College of Science and Technology, Manchester. "Rotating Band and Spinning Band High Vacuum Distillation," by members of the College's Chemical Engineering Department. **6.30 p.m.**

January 29. *North Western Branch.* At Midland Hotel, Manchester. Annual General Meeting and Dinner and Dance. **3.30 p.m. and 7 p.m.**

The Institute of Refrigeration

January 7. At Institute of Marine Engineers, Mark Lane, London, E.C.3. "Some Lubricating Problems of Refrigerating Machines using Refrigerant 12," by H. Heckmatt. **5.30 p.m.**

January 27. At the Savoy Hotel, London, W.C.2. The Annual Dinner.

The Institution of Mechanical Engineers

January 5-7. At 1, Birdcage Walk, London, S.W.1. Symposium on Recent Mechanical Engineering Developments in Automatic Control.

January 27. At 1, Birdcage Walk, London, S.W.1. "Technical and Economic Aspects Covering the Ocean Transportation of Liquid Methane," by Dr. J. J. McMullen. **6 p.m.**

The Institution of Engineering Designers

January 11. At Rutherford College of Technology, Newcastle upon Tyne 1. "Use of Radio-active Isotopes in Industry," by T. V. Suttar. **7.15 p.m.**

The Chemical Society

January 28. At Department of Inorganic and Physical Chemistry, Liverpool University. "Progress in the Study of Heterogeneous Catalysis," by Professor C. Kemball. **5 p.m.**

The Institute of Welding

January 13. At 43 Princes Gate, London, S.W.7. "High Temperature Brazing—A Modern Joining Method," by E. R. Perry. **7 p.m.**

Other Events

January 18-22. Physical Society Exhibition at Royal Horticultural Hall, Westminster, S.W.1.

January 21. At Battersea College of Technology, London, S.W.11. Society of Chemical Industry, Corrosion Group Symposium.

THE CHEMICAL ENGINEER IN WASTE CONTROL

THE chemical engineer enters this field with a dual role to perform. In the first place he is the natural link between the production departments of an enterprise, its management, and the local and national bodies concerned with pollution. In the second place his aim must be to control process losses assigning to this function an importance equal to that conferred upon the more normal aspects of a design process. What is called for in modern industrial conditions is a wholly integrated approach to a project and one which takes into consideration the economy of the complete enterprise. In particular, the possibility of converting the wastes known to accompany processes into useful saleable products should be kept well to the forefront when a process plant is under development.

An industry which offers a fruitful area for the exercise of such an approach is the fermentation industry. In this section of industry which includes the fermentation of molasses, the distillation of grain, brewing, wine manufacture, and, finally, the preparation of antibiotics, a great deal has been done already to make effective use of waste products. In the case of the distillation of grain, what might be termed the waste load has been reduced over the years by a number of measures. Among them is the use of stillage for the manufacture of by-product cattle feedstuffs. And more recently efforts in this branch of the industry have been directed towards the conversion of stillage residues into high vitamin content feeds by fermentation. There is here, therefore, a distinct prospect of a valuable by-product. In the brewing industry wet and dried grains also have been used for feeding purposes, but this does not apply to hop residues which are of little use for this purpose because of their bitter taste; further work, therefore, will be necessary before this waste can become the basis of a useful by-product. Yet another problem in the same branch of industry is how to obtain a greater recovery of dried yeast. The wine industry, too, is beset by some difficult problems which in any case are exacerbated by its seasonal character. For example for every ton of grapes processed something like two hundredweight of wet waste material remains, and so far little has been done to find an outlet for this material.

In the most recent addition to the fermentation family, the antibiotics industry, the integrated outlook has perhaps gained a firmer foothold than in most other members of the family, and recovery systems have been incorporated early in the main manufacturing schemes for the primary products. The industry, too, has used waste treatment extensively as a means of pollution control. However, the fermentation and the different substrates necessary for the production of a wide range of antibiotics confront the engineer concerned with the design of suitable recovery systems with some unusual problems. Since, for example, the fermentations are aerobic there is present the danger of carrying over large volumes of foam to the waste recovery plant. Furthermore, since the yields from antibiotic processes are characteristically low, the ratio of waste to primary product is usually much higher than it is in the older fermentation industries. Also, it must be remembered that each antibiotic has its own peculiar waste recovery problem and that the utilisation of wastes is complicated by the chemical treatment necessary for the separation and purification of the antibiotic. But even though the recovery techniques for this particular field are difficult to devise they are of a kind that can be solved when partly or wholly by the continued application of chemical engineering approaches and methods.

PROFESSIONAL APPOINTMENTS

3 lines 7s. 6d. (minimum), additional word 6d.—Box number 2s. extra.
For Box Numbers reply to BRITISH CHEMICAL ENGINEERING, Drury House, Russell Street, Drury Lane, London, W.C.2.

A VACANCY EXISTS for a qualified Chemical Engineer to undertake research and development work in connection with cryogenic problems. Applications are invited from honours graduates and should be addressed in the first instance to the Secretary, British Oxygen Research and Development Limited, Deer Park Road, London, S.W.19.

SENIOR DRAUGHTSMAN, age 28-35, required for interesting and varied work on the Design and Layout of Plant for the manufacture of Chemicals, must be experienced in the design of pressure vessels and pipework. A non-contributory superannuation scheme is in operation. Applications stating particulars of experience and technical qualifications should be addressed to: The Works Engineer, Midland Silicones Ltd., Barry, Glamorgan.

RESEARCH. A well known Company in the North-West of England engaged in manufacturing building products has an important vacancy in its research department. Applications are invited from graduates under 40 years of age, who have original ideas on industrial processing with an interest in chemical engineering. It is important that candidates should have broad based industrial research and development experience and a capacity for management at a high level. This is a progressive position and conditions of service are very good. Apply giving details of age, education qualifications experience and present salary to Chief Personnel Manager, Box No. B.C.E. 78.

A CHEMICAL ENGINEER is required to work on problems associated with the handling of oxygen and nitrogen. This vacancy provides an opportunity for a Chemical Engineer with a good honours degree to become experienced in this interesting low temperature field of research. Please write to—The Secretary, British Oxygen Research and Development Limited, Deer Park Road, London, S.W.19.

REPRESENTATIVES

WANTED, Representative for correspondence abroad, high earnings, instructions in writing. Write Fortuna Publishing Co., Vienna 1/8, Postfach 49.

TUITION

A.M.I.CHEM.E. More than one-third of the successful candidates since 1944 have been trained by T.I.G.B. All seeking quick promotion in the Chemical and Allied Industries should send for the T.I.G.B. Prospectus. 100 pages of expert advice, details of Guaranteed Home Study Courses for A.M.I.Chem.E., B.Sc.Eng., A.M.I.Mech.E., A.M.I.Prod.E., C. and G., etc., a wide range of Diploma Courses in most branches of Engineering. Send for your copy today—FREE T.I.G.B. (Dept. 43), 29 Wright's Lane, London, W.8.

NORTHAMPTON COLLEGE OF ADVANCED TECHNOLOGY, LONDON
St. John Street, E.C.1.

FOUR YEAR SANDWICH COURSE FOR DIPLOMA IN TECHNOLOGY IN INDUSTRIAL CHEMISTRY

The fifth entry to the Course will be on Monday, 15th February, 1960.

Firms and organisations wishing to nominate students may obtain further details from the College. The College is also able to assist by introducing suitably qualified students to interested firms.

To facilitate industrial staffing arrangements, the Course runs on two entries a year (September and mid-February) so that when one group of students is in College, the other is in industry, the two changing over at six-monthly intervals.

PLANT WANTED

WANTED: Textile Testing Machine, 1,000 lb. capacity. Belting Press, 12 ft. long Gooseneck Press, 2 Daylight, 2 ft. 6 in. long. Mixing Machine for P.V.C. Pastes, 50 lb. capacity. V-Belt Machines. Building, Slicing, Skiving, Slipper Covering, Mould Wrapping and Mould Loading Machines. Steam Vulcanising Pan, 6 ft. long, 4 ft. diameter. Banbury Mixer, 5 lb. capacity. Box BCE 75.

FOR SALE

6 GALLON Staybrite Stainless Steel Containers 16 in. x 11 in. x 11 in., tapering sides, rounded corners, double lid. To clear 30s. each, for quantity, further reduction for large order. 25 gallon S.S. Tanks 28 in. x 8 in. x 30 in. deep, open top. To clear £5 each. P.T.F.E. Rod and Thick Wall Tube, many sizes in stock. L. Solloman Ltd., Beaufort Street, Manchester, 3. BLA 4571.

SERVICE

SHOTBLASTING *in situ* by the "on-site" experts. Organic and inorganic coatings applied. Anything, anywhere, at competitive rates. Darnall Shotblasting Co. Ltd., Doctor Lane, Sheffield, 9. Telephone 42896.

SELL US YOUR SURPLUS STAINLESS STEEL

SHEETS · PLATES · BARS · TUBES
SECTIONS · STRIPS · BLANKS · OFFCUTS

- **top prices paid**
- **offers by return or our buyer will call**
- **prompt collection and payment**

STAINLESS RECOVERIES LTD
Dover Road · Trading Estate · Slough · Bucks
Telephone: Slough 24622

TO OVERSEAS CHEMICAL ENGINEERS AND EXECUTIVES

You can keep in touch with INTERNATIONAL progress in the application of chemical engineering techniques by seeing this leading journal every month.

A subscription costs 50/- *for one year, and only* 100/- *for* 3 *years.*

Please send your order to our Agent in your country, or to HEYWOOD & COMPANY LTD., DRURY HOUSE, RUSSELL STREET, LONDON, W.C.2, ENGLAND.

ARGENTINE

Through Gordon & Gotch Ltd.
Librerias Mackern, Avenue de Leandro N Alem, 457, Buenos Aires.
Mitchells Bookstore, 570 Cangello, Buenos Aires.

AUSTRALIA

Alberts Bookshop Pty. Ltd., 14-22 Forrest Place, Perth.
Angus & Robertson Ltd., 89-95 Castlereagh Street, Sydney, N.S.W.
Centre of Fashion Trade Journals, 100 Flinders Street, (Near Exhibition Street), Melbourne, C.1.
Engineering Publications (Aust.), 52 Pitt Street, Sydney, N.S.W.
H. A. Goddard (Pty.) Ltd., 255A George Street, Sydney, N.S.W.
The Grahame Book Co., 39-49 Martin Place, Sydney, N.S.W.
John Hinton & Co., 50 Miller Street, North Sydney, N.S.W.
Jervis Manton (Pty.) Ltd., 190 Bourke Street, Melbourne, C.1.
J. W. Martin (Pty.) Ltd., Box H561, G.P.O., Perth.
Messrs. Swain & Co. (Pty.) Ltd., Pitt Street, Sydney, New South Wales.
The Tait Book Co. Ltd., 349 Collins Street, Melbourne, C.1, Victoria.
The Technical Book & Magazine Co., 295-299 Swanston Street, (opp. Old Melbourne Hospital), Melbourne, C.1, Victoria.
H. C. Walker, Rich & Co., Grand United Buildings, 149 Castlereagh Street, Sydney, N.S.W.
Through Gordon & Gotch Ltd.
Gordon & Gotch (A'sia) Ltd., 80 Currie Street, Adelaide.
Gordon & Gotch (A'sia) Ltd., G.P.O. Box 1462 T., 262/264 Adelaide Street, Brisbane, Queensland.
Gordon & Gotch (A'sia) Ltd., 511/515 Little Collins Street, Melbourne, C.1, Victoria.
Gordon & Gotch (A'sia) Ltd., 134-144 Parry Street, Perth.
Gordon & Gotch (A'sia) Ltd., G.P.O. Box 1627 B.B., Barrack and Clarence Streets, Sydney, N.S.W.

AUSTRIA

Through Continental Publishers & Distributors Ltd.
Messrs. Morawa & Co., Wollzeile 11, Vienna, 1.

BAHAMAS

Through Gordon & Gotch Ltd.
Moseley's Ltd., P.O. Box 434, Nassau.

BELGIUM

Dohmen Freres, 237 Rue Royale Sainte Marie, Brussels 111 (Schaerbeek).

BRAZIL

Livraria Stark Ltda., Caixa Postal 2786, Sao Paulo.
Through Gordon & Gotch Ltd.
Casa Crashley Ltda., 58, Ouvidor, Caixa Postal 906, Rio de Janeiro.
Publicidade Exletica, S.A., Box 539, Sao Paulo.

BRITISH GUIANA

Mr. Oswald Bentham, General Supplies Agency, Commission Merchants, P.O. Box 375, Georgetown.

BRITISH WEST INDIES

Messrs. Smith & Alfred, 4 Bellsmythe Street, Woodbrook, Port-of-Spain, Trinidad.
Through Gordon & Gotch Ltd.
Stephens & Todd Ltd., 8 Frederick Street, Port-of-Spain, Trinidad.

BURMA

Continental & Overseas Indentors, Post Box 1163, Rangoon.

CANADA

Mr. G. Bray, The Bray Circulation & Advertising, Post Office Box No. 788, Station B, Montreal, P.Q.
The Davis Circulation Agency, Oakville, Ontario.
Wm. Dawson Subscription Service Ltd., 587 Mount Pleasant Road, Toronto, 12, Ontario.
"Periodica", 5090 Ave., Papineau, Montreal 34.

CEYLON

Mr. M. C. C. Wickramanayake, Director, The City Stores Ltd., 48 Ward Street, Kandy.
Through Gordon & Gotch Ltd.
W. E. Bastian & Co., 23 Canal Row, P.O. Box 10, Colombo.
Colombo Apothecaries' Co., Ltd., P.O. Box 31, Price Street, Fort, Colombo.
Plate Ltd., P.O. Box 127, Colombo.

CHILE

Marjan Sucheston, K. Casilla, No. 9950, Santiago.

COLOMBIA

Libreria "El Gato Negro", Carrera 7A, 14-35, Oficinas 201 Y 206, Bogota.

CYPRUS

The Union Trading Co., 41, 41a Zinonos Kitieos Street, P.O. Box 123, Larnaca.
Through Gordon & Gotch Ltd.
S. Ch. Papadopoulos, 88 Ledra Street, Nicosia.

CZECHOSLOVAKIA

Artia Ltd., Ve Smeckach, 30, Praha, 11.

DENMARK

Aksel J. Nielsen, Osterbrogade, 84, Copenhagen O.
Store Nordiske Videnska Bokhandel, Ronersgade 27, Copenhagen.
John Poulsen, The Illustrated News Service, Dahlerupsgade, 1, Copenhagen V.
C. A. Reitzels Bokhandel, Axel Sandal, Norregade 20, Copenhagen.
Victor Schroder, Esq., Hovedvagtsgade, 8, Copenhagen, K.
Th. Wessel & Vett Ltd., Norregade 7, Copenhagen.

ECUADOR

Messrs. "Bravos", Publications Agency, Guayaquil.

EGYPT

The New Publications, Joseph Meshaka & Co., 49 Saad Zaghloul Street, Alexandria, Egypt.
Libraire La Renaissance D'Egypte, Messrs. Hassan & Yousef Mohamed & Co., 9 Sh. Adly, Cairo, Egypt.
Librairie Hachette, 45 bis Rue Champollion, Le Caire, Egypt.
Librairie Hachette, 61 Rue Nabi Daniel, Alexandrie, Egypt.
The New Publications, Joseph Meshaka & Co., 3 Maspero Street, Cairo, Egypt.

FINLAND

Akateeminen Kirkjakauppa, 2, Keskaskatu, Helsinki.
Rautatiekirjakauppa O/Y, Helsinki.

FRANCE

Arthur Reins, 4 Square Moncey, Paris 9e.

GERMANY

American Business Journals, Zaehringerstrasse 28, Berlin-Wilmersdorf.
W. E. Saarbach G.m.b.H., Gertrudenstrasse 30, Koln, 1.
Wm. Dawson & Sons, G.m.b.H., Klosterstrasse 34-36a, Dusseldorf.

GREECE

G. Hagiantoniou, 31 Rue Du Stade, Athens, Greece.

HOLLAND

G. De Vries, Lindenlaan, 10, Zaandijk.

HONG KONG

M. H. Shamdasani & Co., 90B Nathan Road, Kowloon, Hong Kong.
Through Gordon & Gotch Ltd.
The Harris Book Co., 10 Ice House, Street, Kowloon, Hong Kong.

ICELAND

Hilmar Foss, P.O. Box 154, Reykjavik.

INDIA

C. G. Beard, 49 Galstaun Mansions, Park Street, Calcutta.
Khurana Advertising and Publicity Service, 6 The Mall, Agra.
Mahajan Brothers, Opp. Khadia Police Gate, Ahmedabad.
"Manufacturers" Eastern Agency, P.O. Box 301, Bombay.
B. Muni Ratnam, Muny's Subscription Agency, P.B. No. 1653, Madras, 1.
The New Order Book Co., Ellis Bridge, Ahmedabad.
L. V. Paramesh, Mandavali Lane, Mylapore, Madras, 4.
The Popular Book Depot, Grant Road, Bombay, 7.
Mr. B. L. Shah, Magazines Department, Messrs. Charles Lambert & Co., P.O. Box 4087, Bombay, 7.
D. B. Taraporevala Sons & Co., Ltd., "Treasure House of Book", 210 Hornby Road, Fort, Bombay.
The Technical Book Depot, Forjet Terrace, Tordeo, Bombay, 26.
Through Gordon & Gotch Ltd.
Central News Agency, 19/20 Connaught Circus, New Delhi.
New Photoplay Co., 231 Hornby Road, P.O. Box 61, Bombay.
Oxford Book & Stationery Co., Park Street, Calcutta.

IRAN

Through Gordon & Gotch Ltd.
Alfi's Stationery Shop, Avenue Zand, Abadan.

IRAQ

The Empire Trading Co., Khediry Road, Ashar, Basrah.
The Iraq Bookshop, 436A/1 Rashid Street, South Gate, Baghdad.
Through Gordon & Gotch Ltd.
Abdullah Farjo's Bookshop, Ashar, Basrah.

ISRAEL

The Central Subscription Agency, 3, Ben-Yehuda Street, Jerusalem.
Mr. H. Weiss, Weiss' Subscriptions, 22, Allenby Road, Tel Aviv, Israel.

ITALY

Bruno Bollini, Via Verdi, Empoli.
Ulrico Hoepli, Corso Matteotti, 12, Milano.
Interscientia, Corso Garibaldi 34, Venaria, Piemonte.
Editoriale "Maggiora", Piazza 18, Dicembre, 1, Torino 116.
C. Ruggeri & Co., Via Verziere, 2, Milan.
Sperling & Kupfer, Piazza S. Banila, Galleria Metteotti, Milano.
Guido Spiriti, Piazza Risorgimento 10, Milan.

JAPAN

F. Fujita, 586 Kita Senzoku, Otaku, Tokyo.
Fukumatsu Funahashi & Co., No. 11 4-Chome, Kawaramachi, Higashiku, Osaka.
Marusho & Co. Ltd., c/o Messrs. Toyolite & Co. Ltd., 5, 5-chome, Kitonagasa-dori, Ikutaku, Kobe.
Kaigai Shuppan Boeki K.K., 21 Kanda Tsukasacho 2-Chome, Chiyoda-Ku, Tokyo.
Maruzen Co. Ltd., 6 Tori-Nichome Nohonbashi, Tokyo.

JORDAN

Guiseppe Gedeone, Star Street, Bethlehem.

KENYA

Anglo-Tanganyikan News Agency Ltd., P.O. Box 6784, Nairobi.
Jadavji Valji & Sons, P.O. Box 506, Nairobi.
Through Gordon & Gotch Ltd.
East African Standard Ltd., P.O. Box 380, Nairobi.
S. J. Moore Ltd., Government Road, Nairobi.
Nzoia Press Ltd., P.O. Box 34, Kitale
Superior Printing Works, P.O. Box 181, Salin Road, Mombasa.
Warners Bookshop, P.O. Box 36, Nakuru.

LEBANON

Dimitri K. Haddad, P.O. Box 786, Beyrouth.
Elias Andraos Haddad & Freres, P.O. Box 265, Beyrouth.
Gureghian Freres Immeuble-Stephan Rue Fakhreddine, Beyrouth.
J. M. Saad, P.O. Box 352, Beirut.

MALAYA

Davies & Co., Room 22, Bank of China Building, Singapore, 1.
Fongs Trading Co., P.O. Box 220, Singapore.
M. H. Shamdasani & Co., 43 The Arcade, Singapore.
Through Gordon & Gotch Ltd.
Asia Books Co., 91 Brewster Road, Ipoh, Perak.
S. A. Azeez Bros., Railway Station Bookstall, Klang.
The Jubilee Store, 1 and 1a Gladstone Road, Penang.
R. Kandasamy, Railway Station Bookstall, Port Swettenham, Selangar.
Kedah Bookstore, 79 Julan Langgar Alor Star.
Lim Chye Hock, 45 Station Road, P.O. Box 27, Taiping, Perak.
E. K. Yeap & Co., 20 Brewster Road, Ipoh.

MALTA, G.C.

Cachia & Co., 4 St. Anthony Street, Valletta.
Through Continental Publishers & Distributors Ltd.
Sapienza's Library, 26 Kingsway, Valletta.

MAURITIUS

Ebrahim Dawood, Dawood Building, 1 Louis Pasteur Street, P.O. Box 27, Port Louis.
The Mercantile Service Bureau, P.O. Box 262, Port Louis.
Through Gordon & Gotch Ltd.
The Eastern Trading Co., Quay Square, Port Louis.

MEXICO

Sr. Emilio Minon, Apartado 2209, Ave. Arquimedes No. 91, Mexico D.F.

MOROCCO

International Press Agency, P.O. Box 182, Tangier.

NEW ZEALAND

N. Hamilton-Baker, G.P.O. 721, Wellington.
Messrs. R. Hill & Son, Cnr. Crowhurst and Kent Streets, Auckland, S.E.1.
Technical Books Ltd., 48 Grays Avenue, Auckland.
Gordon & Gotch (N.Z.) Ltd., Linchfield Street, Christchurch.
Gordon & Gotch (N.Z.) Ltd., Jamieson Buidings, Dunedin.

NORWAY

Contact Publishing Co. A.S., Akersgt 59, Oslo.
Narvesens Litteratur Tjeneste, P.O. Box 115, Oslo.
Norsk Bokimport A.S., Edv. Stormagt 1, Oslo.
Tryggve Juul Moller (Bokhandel), Fr. Nansens Plass, 8, Oslo.

NYASALAND

Through Gordon & Gotch Ltd.
Blantyre Printing & Publishing Co., P.O. Box 6, Blantyre.

PAKISTAN

Ideal Subscription Agency, 42 The Mall, Lahore.
N. A. Mukhtar & Co., Sukkur, Sind.
Scafar Agencies, Grindlays Bank, Bunder Road, Karachi, 2.

PORTUGAL

Valentim Lourenco, Rua de o Seculo 34-3, o, Lisbon.
A. Vicente Ribeiro, Avenida da Liberdade 236-1°-E. Lisbon.
Through Continental Publishers & Distributors Ltd.
Livraria Bertrand, 73 Rua Garratt 75, Apartado N. 78, Lisbon.

SOUTH AFRICA

British Technical Journals, 15 Baines House, Broad Street, Durban.
Lindex (South Africa) Pty. Ltd., P.O. Box 4536, Johannesburg.
Central News Agency Ltd., 58 Maidland Street, Bloemfontein, Orange Free State.
Central News Agency Ltd., P.O. Box 599, 60/62 Oxford Street, East London, Cape Province.
Central News Agency Ltd., P.O. Box 356, 16 Adderley Street, Port Elizabeth, Cape Province.
Central News Agency Ltd., 86 Visagie Street, Pretoria, Transvaal.

SOUTHERN RHODESIA

Messrs. Felix Sutherland (Rhodesia) Ltd., 20 Orr Street, Salisbury.
Mr. V. W. Van Rijnberk, P.O. Box 238, Bulawayo.
Kingstons Ltd., P.O. Box 256, Cnr. Fife Street and Sixth Avenue, Bulawayo.
Kingstons Ltd., 89 Victoria Street, Salisbury.

SPAIN

J. Diaz de Santos, Libreria J. Diaz de Santos, Calle Lagasca 38, Madrid.

SUDAN

Through Gordon & Gotch Ltd.
The Sudan Bookshop, P.O. Box 156 Khartoum.

SWEDEN

J. Ekelund, Knislinge.
Stig. Falk, Vallingatan, 13, Stockholm, 1.
A. B. Nordiska Kompaniet, Hamngatan 18-20, Stockholm, 7.
A. B. Siowalls Bokhandel, Drottninggatan 86, Stockholm.
Wennergren-Williams, A. B., Fack, Stockholm, 30.

SWITZERLAND

Mr. Willi Hardi Grossisteh No. 100 571, Ober-Muhen.
W. Frey-Schneeberger, Mattenweg 13, Lenzburg.

SYRIA

Vahan Darakjian 291, Atfaiyeh St. Aleppo, Syria.

TANGANYIKA

Through Gordon & Gotch Ltd.
Arusha Bookshop, P.O. Box 246 Arusha.

THAILAND

Teck Heng & Co., 1326 New Road, Bangkok.
The Sales Agency Co., 1091/1093 New Road, Bangkok.

TUNISIA

Maison Allal, Allal Emma, 18 rue Durand-Claye, Tunis.

TURKEY

Bedros Kapman, P.O. Box 2158, Istanbul.

URUGUAY

Kurt Palnitzsky, Casilla Correo 293, Montevideo.
R. Raphael, Convencion 1488, Montevideo.

U.S.A.

The Overseas Publishers' Representatives, 29 West 34th Street, New York, N.Y., U.S.A.

British Chemical Engineering
A HEYWOOD PUBLICATION

Low-temperature Heat Transmission

TOWARDS the end of the nineteenth century the celebrated scientist D'Arsonval forecast that the successful liquefaction of air and the separation by distillation of its two main constituents would revolutionise not only agriculture but also the gas, chemical and metallurgical industries. Even a casual look today at these industries is enough to show how close to the mark this forecast was. Tonnage oxygen, for example, is widely used in the steel and iron making industries, where it is the most important single factor in achieving greater output from existing equipment; it is essential to at least one process for the manufacture of that basic chemical acetylene, as well as to a number of important gasification processes. And several important processes for the manufacture of nitrogenous fertilisers depend upon the supply of large quantities of nitrogen derived from the distillation of air.

Here we may observe that we are nowadays so accustomed to a plentiful supply of nitrogenous chemicals that we apt to forget that in the past the position has not always been so secure; for in the era to which D'Arsonval belonged, a number of voices, including that of Sir William Crookes, had predicted a nitrogen famine. That this gloomy prediction proved false but was in fact the direct opposite of the situation which eventually emerged is testimony to D'Arsonval's sagacity: it also emphasises the extent of our indebtedness to those early pioneers such as Claude, Linde and Hampson and those others who laid the foundation stones of modern low-temperature technology. Their efforts have paved the way for such developments as the production of ethylene, the large-scale separation of helium, the distillation of the isotopes of hydrogen, and, as a consequence of the tonnage production of oxygen and nitrogen, the extensive output of the rarer gases such as argon with its important role in the sphere of plant fabrication.

In all such processes the low-temperature heat exchanger performs a vitally important function and it was this equipment which put the finishing touch to the early continuous processes for the liquefaction of the so-called permanent gases. Its practice probably began more than a century ago when Siemens attempted to liquefy air. The machine he used was essentially a simple compressed-air motor combined with a concentric-tube exchanger, which allowed the outgoing expanded air to cool the incoming compressed air. As the saying goes "there is many a slip twixt cup and lip", and this apparently sound process which attracted the attention of a number of investigators, Hampson included, did not liquefy air. The principal reasons for this lack of success were the congelation of the engine lubricant, the loss of refrigerant through poor thermal insulation, and the accumulation of impurities in the system. The lessons of this failure were learned and Hampson, by replacing the engine by a throttle, removing the carbon dioxide from the air and retaining the exchanger, albeit in a much improved form, succeeded in producing liquid air. Hampson's multi-tubular, coiled cross-flow countercurrent exchanger may be justly regarded as the forerunner of the modern low-temperature heat exchanger.

The design problems of the low-temperature heat exchanger differ considerably from those of the normal process plant exchanger. Thus in order to obtain a close approach to thermodynamic reversibility—a necessity if the process is to be economic—the designer must use small temperature differences which in turn mean large surfaces. Yet at the same time he must avoid serious pressure losses; thus in order to conserve refrigeration he must make the exchanger as compact as possible. The ways in which designers have sought to attain these ends have changed considerably with the passage of time. To appreciate this point we need only compare the earlier Linde design of exchanger, which was an iron affair some 300 ft or more in length, with the elegant plate-fin exchangers described by Mrs Lenfestey in this issue (see p. 27). Another type of exchanger to receive attention in this issue is the regenerator. Developed in theory by Hausen, its introduction in the mid-'twenties by Frankl was one of the milestones not only in low-temperature heat-transfer practice but in gas separation processing generally. For this device, which is more efficient, cheaper and more compact than the tubular varieties of exchanger, is indispensable to the tonnage oxygen plant.

Undoubtedly, of the professional bodies in Britain with an interest in low-temperature technology, including its heat-transfer practice, the outstanding role of development has been played by the Low Temperature Group of the Physical Society. This Group, which was formed towards the end of 1945 under the chairmanship of Sir Alfred Egerton, devoted its inaugural meeting to the technical problems of gas separation, the relevant paper being delivered by A. M. Clark. Since that time the various papers given under its auspices, including those presented at symposia organised jointly with other bodies, including the Institution of Chemical Engineers, amount to a very impressive list. The subjects covered have ranged from the essentially engineering aspects, including the production and use of gaseous oxygen, expansion turbines, distillation, automatic control and instrumentation, to facets of the field more remote from engineering practice. In this category we recall with pleasure a fascinating paper given by the late Sir Francis Simon on the unattainability of absolute zero.

Among the symposia of distinct chemical engineering flavour we would mention two in particular. The first of these, held in conjunction with the Institution of Chemical Engineers late in 1957, was concerned with developments in industrial gas separation processes, and the second, a single-handed effort held towards the end of last year, was devoted to low-temperature heat transfer. Indeed, this symposium has provided the main contents of this the first issue of BRITISH CHEMICAL ENGINEERING for the year 1960.

66.045: 1-974

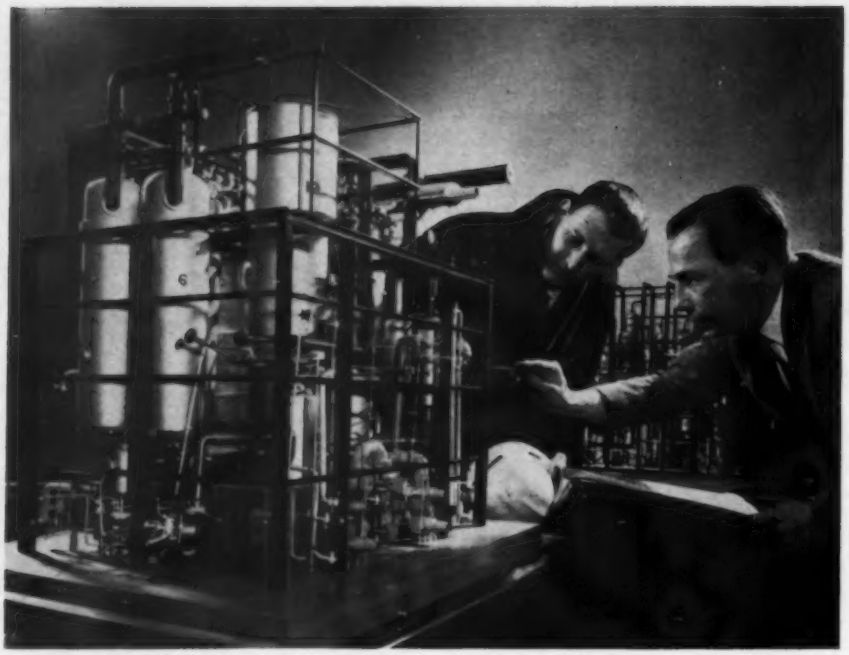


Fig. 1. Scale model of the air separation plant of the Shell Haven Refinery. Models such as these proved invaluable during construction work.

LOW-TEMPERATURE HEAT EXCHANGERS

The available types, their special functions, and importance to the economy of gas separation processes are outlined

by P. M. SCHUFTAN, Ph.D., M.I.Chem.E.

IN this introductory paper I will review the special requirements and features of low-temperature heat exchangers from the experience gained by my company in a great number of plants over many years. When we talk today about low-temperature heat exchangers we have in mind heat exchangers as applied to low-temperature gas separation processes and specifically air separation processes. Very large heat exchangers can be required for this purpose; for instance, the largest air separation plant operating in this country, producing about 250 tons/day of oxygen, requires for the heat exchange between air and the waste nitrogen an exchanger with a throughput of about 1000 tons of air per day, i.e., a size which will hardly be exceeded by exchangers used at higher temperature levels in the chemical industry.

I will first deal with general features of L.T. heat exchangers and then with the applicability of different types for particular duties.

Economics of heat exchangers are determined by the fixed costs of the exchanger, i.e., costs for materials and labour as inherent in the particular configuration of the heat exchange surface, and by the energy consumption determined by the efficiency of the exchanger which is measured by temperature difference and pressure drop, both being a function of the heat-transfer coefficient. Obviously, the sum of fixed and running costs should be a minimum in each particular case.

For low-temperature heat exchangers, the choice of *materials* is somewhat simplified from a *chemical* point of view, as corrosion difficulties can generally be ignored, the exception being the few cases where the gas mixture to be separated contains constituents which are liable to attack metals in the absence of liquid water, and which can either not be removed by chemical purification, or have to be recovered as such by condensation. However, *mechanically* there are limitations due to brittleness of certain metals at low temperatures, and it is necessary that the metal is easy to work and able to produce a gas-tight exchanger; in certain cases differential thermal contraction will have to be considered if the exchanger consists of more than one metal. In practice, only copper, aluminium and austenitic steel as such or as alloys, have been found to be suitable for low-temperature work.

In the case of high-temperature exchangers the heat-transfer surface to be provided has normally to contain a *fouling* allowance to cater for deposits formed by chemical reactions, such as polymerisation or scaling-up, or rusting from the cooling water side. No such allowance should be necessary for low-temperature exchangers if

Dr. Schuftan *is with The British Oxygen Engineering Co. Ltd., London, E.*18.

care is taken that the gas mixture to be cooled does not contain solid or liquid suspensions or trace constituents liable to create deposits by chemical action. If these conditions are met, solid deposits can only arise by condensation and can then be regarded as *reversible*, as in view of their high vapour pressure, sometimes combined with a low melting point, they can be completely removed in a batch operation.

Accumulation of Solid Condensates

It is remarkable that solid condensates, especially water, CO_2 and benzole, do not impede the heat transfer as orientated growth as crystals perpendicular to the heat-transfer surface seems to occur. The fins thus formed have certainly a low efficiency, but provide additional heat-transfer surface. Thus solid condensates can be allowed to accumulate until the pressure drop of the heat exchanger becomes too high. However, where gas constituents *condense as liquids*, such liquids should not, during the further cooling process, be allowed to be carried into a zone where they could freeze, as otherwise a rapid blockage of the heat exchanger would result, preceded by deterioration of the heat transfer. Vertical heat exchangers, designed to enable reflux of the condensed liquid against the ascending gas stream, are one means of avoiding this difficulty.

The particular conditions of gas separation processes require mainly a gas-to-gas heat exchange, whereby there is always a marked *pressure difference* between the process gas and the separation products. Thus, the process gas section of continuous heat exchangers can be designed for a considerably higher mass flow rate than the low-pressure section without exceeding the permissible pressure drop.

The exceptionally stringent requirement of *leak tightness* of low-temperature heat exchangers is, perhaps, not always fully appreciated. For instance, in an air separation plant producing 10,000 cu. ft/hr of oxygen at 99.5% purity and operating at an air pressure of 1000 psig, a pin hole of 6 thou. diameter would be sufficient to deteriorate the oxygen purity to 99.0% and thus make the oxygen unsuitable for cutting and welding. In a plant producing 100 tons of oxygen/day, and operating at a pressure of 72 psig, a hole of 1 mm diameter between the air and the oxygen sections, or regenerator valves having an equivalent leakage, would deteriorate the oxygen purity by 0.1%.

When producing pure nitrogen containing 10 ppm of oxygen a leak of only 0.003% of the process air into the nitrogen doubles its oxygen content. Similar conditions apply in so-called "nitrogen wash" plants where a practically complete elimination of carbon monoxide and other contaminants from ammonia synthesis gas is required.

What effect has the *efficiency* of low-temperature exchangers on the power consumption of the process? The cold loss resulting from the warm end temperature difference has in practice to be covered at the lowest operating temperature and according to the second law of thermodynamics, and assuming an efficiency of 50%, about 150 k. cal. can be produced per kWh at 77°K. This means that in a tonnage oxygen plant each additional °C of the warm end temperature difference increases the power consumption of the air separation unit by about 2%.

In view of the different shape of the *enthalpy/temperature* curves of the compressed process gas and the low-pressure separation products, the mean temperature difference of low-temperature heat exchangers is in many cases not determined by the warm and cold end temperature differences as derived from the mass and cold balance, but a minimum temperature difference may occur at an intermediate point. In this case excessive cold losses at the warm end can only be avoided by suitable adjustment of the mass ratios of the two gas streams, either over the whole or over part of the length of the exchangers.

An *increase in the pressure drop* of the separation products in heat exchangers increases the operating pressure required for the process gas. For tonnage oxygen plants each additional lb. of pressure drop increases the power consumption of the air separation unit by about 2%. In addition, it makes the sublimation of condensates in regenerators or reversing exchangers more difficult and reduces the cold production of the expansion turbine.

Low-pressure drop and high heat-transfer rate are rather conflicting requirements and especially in large plants it is important to find the economical balance which is appropriate to the particular conditions.

Coming now to the *applicability* of different types of low-temperature heat exchangers for particular duties, *straight tubular exchangers* are of special interest as reflux coolers or reflux condensers. *Concentric tubes* which can be wound in coils are still used in a few cases, the inner tube or tubes carrying high-pressure gas while the outer space is used for the low-pressure gas.

More efficient is a *cross-flow*, counter current type, a design first developed by Hampson for his air liquefier and now especially used in L.O. plants. It consists of a cylindrical shell with central closed mandrel, carrying layers of high-pressure tubes with alternative direction of the winding. The parallel tubes are joined in inlet and outlet headers. The number of tubes in each layer and the axial and radial spacing are so chosen that each tube has the same length and this ensures uniform distribution of the H.P. gas through all tubes. The L.P. gas flows in countercurrent in the space between the tubes and the shell. Heat-transfer conditions in such exchangers are well known from studies of various investigators. In view of the coil configuration of the tube bundle, no special devices are required to cater for differential thermal contraction or expansion.

For medium or low pressures *non-tubular exchangers* (a classification into which regenerators also fall) have become important.

Introduction of Plate and Fin Exchangers

Plate and fin-type exchangers have only relatively recently been introduced into the low-temperature technique. They consist of corrugated sheets sandwiched between flat sheets and brazed together, and they are used not only for heat exchange between gas and gas as continuous or reversing exchangers, but also between gas and liquid (undercoolers) and as condenser/reboilers.

Regenerators exchange heat from gas to gas through the intermediary of a heat storing mass. The pair of vessels are filled with suitable packing and after a period determined by the heat capacity of the packing, the vessels are changed over automatically by time or temperature impulse actuating valves at the warm ends, while automatic check valves are provided at the cold ends. Regenerators, which were first used for low-temperature heat exchange by Fränkl, are characterised by low-pressure drop and low heat-transfer coefficients, but the packing provides a very large heat-transfer surface. The original corrugated aluminium strips used by Fränkl have now been largely superseded by stone filling which has a higher mass to surface ratio, resulting in longer cycle periods. Such stone filling allows the provision of coils for indirect heat transfer within the regenerators which makes it possible to produce separation products of high purity and uncontaminated by either process gas, water vapour or CO_2.

An unavoidable disadvantage of regenerators is the loss in process gas occurring when changing over from the vessel under pressure, to the vessel which is at low pressure. This loss can be reduced by an intermediate pressure-

equalising step between the two vessels, but will even then amount to about 3½% of the process gas in the case of tonnage oxygen plants.

Here I will only briefly mention the important function of regenerators as very efficient *purifiers*, as they remove by condensation on the packing gas constituents of high freezing point which can be resublimated by a part or the whole of the separation products. In order to ensure complete revaporisation, it is, however, necessary to watch the actual volume ratio of the two gases which are in heat exchange with each other, their temperature difference along the length of the regenerators and the vapour pressures of the deposits at their condensation and sublimation temperatures.

It is well known that this condition of complete sublimation is not automatically met and certain "balancing" methods have to be used to create favourable temperature differences at the critical parts of the regenerator. Generally, the volume ratio of return gas to process gas in the colder part of the regenerator will have to be increased, either by using the so-called "Trumpler Pass" or by bleeding off at an intermediate point of the regenerators part of the process gas, and cool it separately by other means.

Reversing exchangers, which have been developed by Professor Collins of the M.I.T., differ from regenerators in that they are not heat storers, nor do they need to be provided in duplicate. As in normal exchangers, the transfer from gas to gas takes place through a metal wall. When the process gas section begins to block with ice or carbon dioxide, the flow through the process and return gas sections is automatically reversed. The cross-sectional areas of the two sections must thus be kept equal and this means that either the heat-transfer rate on the process gas section will be low or the pressure drop on the return gas section high. This is a disadvantage of this type of exchanger. Balancing is just as necessary as on regenerators and the preferred method is the use of the Trumpler Pass, whereby the balancing air is rewarmed in a special pass of the exchanger, which can easily be provided.

While reversing exchangers were originally designed as concentric tubes with fins in the interspace, plate and fin exchangers are now generally used. In large plants it is not possible to accommodate the whole heat-transfer surface required in a single unit and several exchangers have thus to be arranged in parallel, which involves certain distribution problems.

An essential element in low-temperature gas separation processes is the *condenser reboiler*. The condensating fluid is practically always the gas mixture to be separated or at least, a component of this mixture; i.e., in the case of air separation it is air or nitrogen.

The following conditions have to be met:

(1) The condensation must be complete in order to obtain adequate reboil and reflux with the minimum amount of heating fluid.

(2) The temperature difference between the condensing and boiling fluids must be as small as possible; e.g., in tonnage oxygen plants the operating pressure of the air is controlled essentially by the temperature difference in the condenser-reboiler. Starting from a boiling temperature of the liquid oxygen of 93°K and increasing the temperature difference from 2°C to 3° and 4°C, the condensation pressure rises from 5.5 to 5.9 and 6.4 kg/cm² absolute, and this increases the power consumption of the plant by 4.2 and 8.9% respectively, which is quite serious.

(3) The accumulation of incondensible constituents on the condensing side must be avoided by venting in order to obtain satisfactory heat transfer.

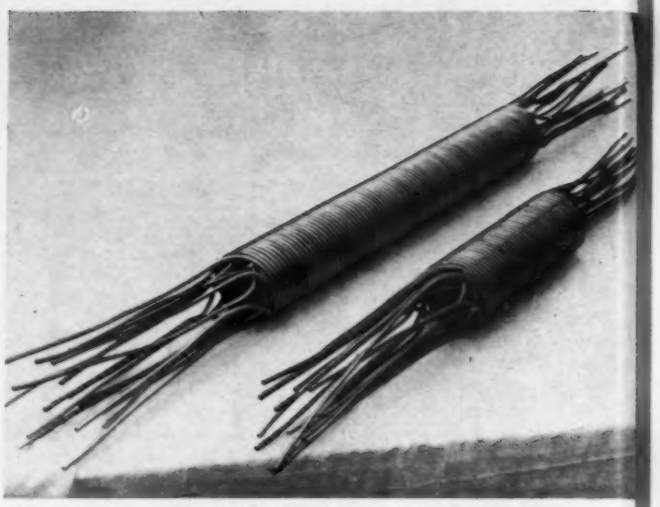


Fig. 2. Heat exchanger (top) and pre-cooler (bottom) of the cross-current type, high-pressure coils wound on mandrel ready for threading into headers.

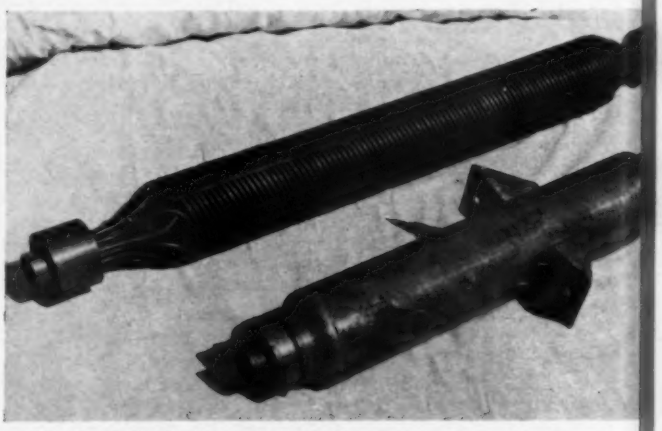


Fig. 3. A finished pre-cooler and interior of a main heat exchanger for a liquid nitrogen plant. (Photos: Refinery Equipment & Speciality Co. Ltd.)

(4) The condensate must be effectively drained to avoid blanketing of the heat-transfer surface by liquid.

(5) On the boiling side vapour bubbles must be removed from the heat-transfer surface as quickly as possible, in order to utilise it fully. This can be achieved either by providing a relatively large cross-sectional area on the boiling side or by ensuring quick circulation of the liquid to be vaporised.

In some cases condenser-reboilers consist of coils of one or more parallel tubes. Another type widely used in double column air separation plants consists of a bundle of tubes arranged vertically between two tube plates with the upper one as a floating head. The velocity of the condensing nitrogen is kept low enough to enable it to reflux into the lower column. For a 100 ton oxygen plant this condenser type requires not less than 14,000 tubes.

Measurement of the heat-transfer coefficient on condenser-reboilers is not easy and especially difficult in the case of partial condensation such as, for example methane from coke-oven gas.

662.995-974 : 66.045.1-974

THE APPLICATION OF REGENERATORS AND HEAT EXCHANGERS*

The comparative merits of reversing heat exchangers and regenerators for low-temperature heat-transfer duties

by D. E. WARD, Ph.D., A.M.I.Chem.E.†

IN low-temperature plant the feed gas must be cooled to the operating temperature, usually the dewpoint of the gas, by the cold outgoing product and waste gases. This requires heat exchangers which are not only efficient but possess an inherent "self-purging" action, since in most cases the feed gas will contain impurities which deposit out on the exchanger surfaces as the feed is cooled. Two types of heat exchanger which are particularly suited to this duty are the regenerator and the reversing exchanger.

The essential heat-exchange requirements of whatever type of equipment is used are:

(1) the pressure drop must be small to conserve power;
(2) the exchanger must be able to operate at small temperature differences, which can only be obtained if serious maldistribution can be avoided, to conserve refrigeration and the associated power;
(3) the exchangers should be compact, i.e., possess a large surface/volume ratio; and
(4) the capital cost must be low.

The self-purging action to remove deposited feed gas impurities can be achieved by the periodic switching of the feed and return gas channels, so that the impurity deposited by the feed gas is evaporated by the colder return gas. The colder gas is able to evaporate all the deposited impurity because of what may be termed the "pressure effect". The feed gas enters the plant under pressure, whilst the return gas leaving the plant is usually at almost atmospheric pressure and is colder, but due to its lower pressure its saturation concentration is higher and the mass transfer driving forces are higher. This enables all the deposited solid to be evaporated, provided the temperature difference between the feed and return gas is at no point allowed to exceed the value given approximately by the equation.[1]

$$\Delta T < \frac{2 \cdot T_s^2 \left\{ \frac{C_s^{*L}}{C_s^{*H}} - 1 \right\}}{\varepsilon \left[(C_s^{*L}/C_s^{*H}) B^H + B^L \right]} \qquad \ldots . (1)$$

where T_s = absolute surface temperature;
ΔT = maximum allowable overall temperature difference assuming no wall resistance;
ε = ratio of heat transfer to mass transfer rate (1.0 for water/air and 1.2 for CO_2/air system);
C_s^{*L} = equilibrium concentration in the high-pres- gas at temperature T_s;
C_s^{*H} = equilibrium concentration in the high-pressure gas at temperature T_s; and
B^H and B^L = the constants in the equation $C = e^{A - \frac{B}{T}}$ expressing the equilibrium gas concentration in the high- and low-pressure streams respectively as a function of temperature. The equation only applies for equal molar feed and return gas flows. Fig. 1 gives some typical values of ΔT for the CO_2/H_2 and N_2/H_2 system,[1] calculated from Equation (1).

The switching time required to prevent excessive increase of pressure drop by deposited solid can be estimated by the equation[1]:

$$m = \left(\frac{M_l}{M} \right) \frac{G \cdot C_s^{*H} \cdot C \cdot \mathrm{Re}^{-a} \cdot \mathrm{Sc}^{-b}}{(T_s^2/\varepsilon \cdot \Delta t \cdot B^H) - 1} \qquad \ldots . (2)$$

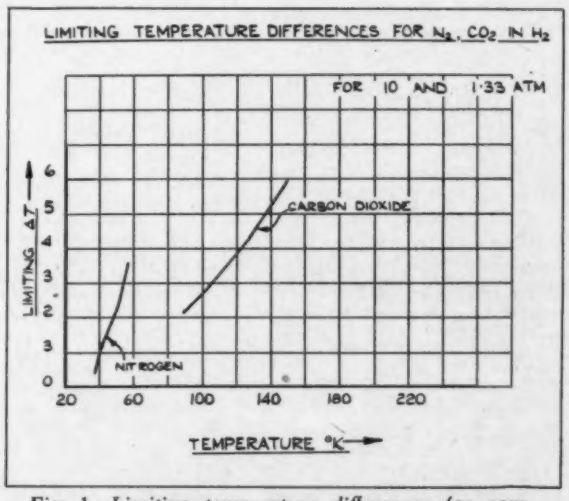


Fig. 1. Limiting temperature differences for complete evaporation of deposited solids for the systems CO_2/H_2 and for N_2/H_2.

* Synopsis of a talk given to the Low Temperature Group of the Physical Society at a one-day discussion on Low-temperature Heat Exchangers, Tuesday, December 9, 1958.

† British Oxygen Engineering Co., London, N.18.

where m = mass deposition rate;
M_1 = molecular weight of impurity;
M = molecular weight of bulk gas;
G = mass flow rate;
$\triangle t$ = temperature difference between surface and bulk of gas and C, a and b are the constants in the heat-transfer relation; and
Sc = Schmidt number

$$St = C \cdot Re^{-a} \cdot Pr^{-b}$$

where St = Stanton number;
Re = Reynolds number; and
Pr = Prandtl number.

In some non-ideal cases[2] the "pressure effect" is reduced and can be even reversed so that the equilibrium concentration of the high-pressure gas is greater than the low-pressure gas at the same temperature. This effect usually occurs when the bulk gas approaches its critical temperature. If the effect is reversed, then regenerators and reversing exchangers will not remove the deposited solid and absorption, chemical purification, blocking exchangers or some other method must be employed.

To ensure re-evaporation, important features of the heat-exchange equipment are:

(1) the ability to control local temperature differences;
(2) identical high- and low-pressure channels so that the flow of return gas is the same through both channels; and
(3) the packing or extended surface should have some filtering action to discourage solid impurities from being blown to the colder regions from which they cannot be evaporated.

Precautions must also be taken to ensure that any oil droplets entrained in the feed gas are removed before entering the low-temperature plant.

Regenerator

A regenerator is a heat-storage device and usually consists of two cylindrical vessels packed with metal ribbon (Frankl packing) or stones, and the warm feed gas and

TABLE I.—COMPARISON OF REGENERATORS AND REVERSING HEAT EXCHANGERS

Property	*Regenerator*	*Reversing Exchanger*
Heat-transfer area per unit volume	High, usually about 500 ft²/ft³. Ordinary shell and tube about 50 ft²/ft³.	Plate fin. 200 ft²/ft³.
Volume of exchanger	Fairly large volume due to need for heat storage. Larger power loss on switch-over. Greater contamination of return gas or larger purge loss. Greater loss of feed to process.	Very compact. More likely to become blocked because of the smaller volume in which the solids can collect.
Weight	The filling material is a considerable weight.	Light—use very thin extended surface.
Maximum operating pressure	No theoretical limit. Practical limit change over loss. Usually not more than about 6 atm.	Plate fin up to 20 atm.—caution required because of fatigue failure due to constant reversals. Concentric tube can be used above 20 atm.
Shape	Length/cross-section ratio usually good.	Smaller surface area/unit volume. Tendency for long, thin shape.
Maldistribution	Can take large gas flows in one unit. Serious maldistribution unlikely to occur if carefully packed.	Plate fin and concentric tube of limited cross-section, must use several units in parallel. Care needed to avoid maldistribution and associated large temperature differences, preventing complete evaporation of solid.
Cycling of outlet temperature	Varies over a fairly wide range.	Constant exit temperature except at moment of reversal.
Maximum concentration of feed impurity which can be handled	Due to the larger volume and the continuous shift of packing temperature during each cycle, a much higher concentration of feed impurity can be handled. Have been used for up to 100 times CO_2 content of air (i.e., 3% CO_2). (Regenerators for coke oven gas.)	The space around the secondary surface is limited and the solid is deposited along the length according to the vapour pressure curve, and so the maximum concentration of feed impurity which can be handled is lower.
The ability to exchange heat between more than two gas streams without contamination by the feed gas or evaporating impurities	Coils must be embedded in the packing. This is costly and the throughput is limited. If no coils are used, contamination of return product gas by mixing with feed gas and evaporating impurities. Mixing with feed gas can be eliminated by suffering a purge loss. Contamination by evaporating impurities cannot be avoided. Local temperature differences cannot be controlled except by use of coils or bleeding regenerator.	Permits a pure product to be obtained. No purge loss. Permits control of local temperature differences by recirculation.
Cost	Packing cheap—main cost is for shells. Very suitable for large-scale low-temperature plants.	More expensive than regenerators, but gains due to pure product, etc., may outweigh this.

cold return gas are switched alternately between the vessels, so the heat of the feed gas stored in the packing is used to heat the return gas to room temperature and vice versa. At the warm end the gas streams are switched by pneumatically-operated valves through a timing mechanism and at the cold end by check valves, which open or close according to the direction and force of the gas pressure exerted on them.

Reversing Exchanger

A line diagram of the operation of a reversing exchanger is shown in Fig. 2. Pneumatically-operated valves are used to switch the gas streams at the warm end and check valves at the cold end. A high-pressure recirculation system is shown for the control of local temperature differences. A low-pressure system can be used instead, but this increases the low-pressure, side-pressure drop which is not desirable and requires an extra small exchanger. The two most essential requirements of a reversing exchanger are good thermal contact between gas streams and an extended surface in the gas channels which, besides increasing the heat transfer, has a filtering action preventing the blowing down of solid impurity to colder parts from which it cannot be evaporated.

Only two types of exchanger have so far been developed which are suitable for reversing exchanger duty, the concentric tube and the plate fin type.

The concentric tube type with helical coil surface extension was developed during World War II by PROFESSOR S. C. COLLINS[3] for portable oxygen units and is only suited to small gas flows. The plate fin exchanger[4] consists of parallel plates of aluminium alloy with extended surface between the plates to form a block which is brazed together in a molten salt bath. These exchangers can handle fairly large gas flows, but at the present time the maximum cross-section of each block is limited to about 2 sq. ft and the length to about 9 ft, so that for the larger plant, e.g., tonnage oxygen, several blocks in parallel and series are needed.

The relative merits of reversing heat exchangers and regenerators are given in Table I, which is presented on the preceding page.

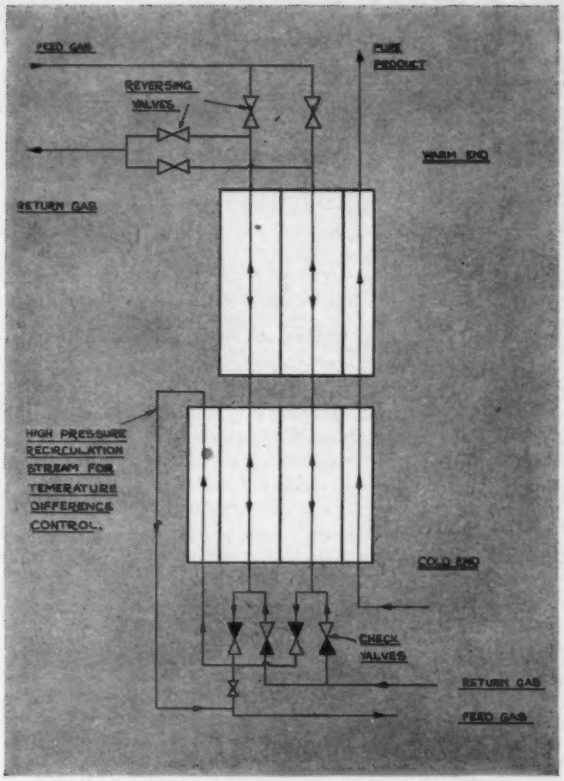


Fig. 2. Diagram showing the method of operating reversing exchangers.

REFERENCES

[1] Denton, W. H., Shaw, B., and Ward, D. E. "The Purification of Hydrogen for Distillation", *Trans. Inst. Chem. Eng.*, 1958, **36**, No. 3, 179-200.

[2] Webster, T. J. "The Influence of Pressure on the Equilibrium between Carbon Dioxide and Air", *Proc. Roy. Soc. A.*, 1952, 214, 61.

[3] Lobo, W. E., and Skaperdas, G. T. "Air Purification in the Reversing Exchanger", *Trans. Amer. Inst. Chem. Eng.*, **43**, No. 2, 69-73.

[4] Denton, W. H., and Ward, D. E. "Applications of Plate-fin Heat Exchangers". (See p. 18 this issue.)

Reddish Chemical's New Factory

Provided with fully equipped research laboratory

The Reddish Chemical Co. Ltd. have recently opened new works and offices at Stanley Road, Cheadle Hulme, near Stockport. They have been specialists in the manufacture of chemical detergents for 30 years, supplying to the milk, food and beverage industries, the engineering trades and to farming. With these activities, coupled with close liaison with ministerial groups and their respective research establishments, Reddish have secured for themselves a high reputation for their products.

The new works and offices have been built on a 3½-acre site in the recently developed industrial area at Cheadle Hulme. The factory itself has a floor space of 16,500 sq. ft with a separate office block of more than 2000 sq ft. The new Cheadle Hulme site allows for considerable further expansion of works and offices, as well as storage facilities and lorry accommodation. The factory floor is at two levels to make full use of gravity. Raw materials are brought in at one end and elevated mechanically to the higher level, where they are at the right height for being fed into the various mixing plant. The mixed material is discharged at the lower floor level, packed and stored a short distance away ready for dispatch at the lorry-loading bay. The lower floor level is itself raised 4 ft above ground level so that the product is packed and stored at lorry level, reducing manhandling to a minimum. The factory is divided into two sections: that which deals with the colloidal and liquid side of manufacture, and that which is concerned with the dry powder. At the extreme end away from the processing plant, and separated from the main building, empty returned drums are cleaned under high-pressure steam jets.

A vitally important aspect of the company's activities is research, both for quality control during manufacture and for product application in new and existing fields, and for this purpose a fully equipped laboratory has been provided within the factory building.

The factory is steel-framed and constructed of brick with a floor of reinforced concrete. Steel-framed side windows and northern lights let into the asbestos roof give ample light, and a dust extraction plant provides clean air and favourable working conditions. One of the most interesting features of the factory is the attention which has been paid to the roadways and loading areas and the best possible use has been made of a one-way road system.

66.045.1:66.074.8

APPLICATIONS AND DESIGN OF PLATE-FIN HEAT EXCHANGERS

A method of designing these compact heat exchangers is described together with examples of their application

by W. H. DENTON, B.Sc., F.Inst.P., and D. E. WARD, B.Sc., Ph.D., A.M.I.Chem.E.

THE choice of the most suitable type of heat exchangers for a process and their design for a required balance between their capital cost and the energy consumption they save is a standard problem. It is more important in low-temperature gas separation plant than in high-temperature processes, since most of the plant consists of gas/gas heat exchangers and there is a much greater economic incentive to use large heat exchangers with high thermal efficiencies. Also, the exchangers handling the feed gases have to be suitable for low-temperature gas purification.

The plate-fin gas/gas heat exchanger is the most recently developed type for low-temperature plant and is now being used more extensively in air separation plants. A recent design study of a plant for the large-scale distillation of hydrogen at 20°K for the production of deuterium, supported by experimental work,[1, 2] concluded that plate-fin units should satisfy all the requirements of the process and for certain duties had many advantages over alternative types. Also, with the exception of packed regenerators for some duties, they were the cheapest. This paper includes the design methods developed for the plate-fin heat exchangers in the above work.

Requirements of Low-temperature Plant Heat Exchangers

The need for high thermal efficiencies can be appreciated as follows. In a process operating at a high temperature, a heat exchanger simply reduces the external *heat* to be supplied at the high temperature to an acceptable value, Q, which is proportional to the heat exchanger temperature difference and is rejected at the temperature of the surroundings, T_0°K. On the other hand, in a process operating at a low temperature, T_1, below T_0, the same amount of heat, Q, which flows from T_0 to T_1 requires the expenditure of work to absorb the heat at T_1 and reject it at T_0. Even with a thermodynamically-ideal refrigeration cycle, this work is:

$$Q \cdot \left(\frac{T_0}{T_1} - 1\right)$$

which, for an air separation plant, for example ($T_0 = 290$°K, $T_1 = 80$°K), is $2.6Q$ (work), while the cost of power is nearly four times that of the equivalent heat at high temperatures. Thus the energy cost, and therefore the incentive to use a higher thermal efficiency (lower temperature difference), is larger for the low-temperature process at 80°K by a factor of at least 10.

These high efficiencies have to be obtained within the usual limits set by the additional power consumption to overcome pressure drop in relatively low-pressure gas streams, by capital cost and by the increasing importance of ensuring uniform distribution of gas flow (in and between units in parallel) for the high ratios of temperature span to temperature difference used in low-temperature plant.

Compactness of the heat exchanger is also very important because of the expense of the thermal insulation to reduce external heat inleakage and the consequent refrigeration load. Thus a large surface area/volume ratio is required.

Another critical requirement is that the heat exchangers used to cool the feed gases must be suitable for gas purification, i.e., all the solid deposits of impurities condensed on their surfaces must be continuously re-evaporated by operating them as reversing or alternating dual units, the principles of which have been described previously.[1] This is the most critical problem in low-temperature gas separation processes. Apart from imposing certain limiting maximum temperature differences, it requires that the impurities must immediately condense as solid deposits on the heat exchanger surfaces and not blow down as solid particles towards the cold end of the heat exchanger and block it, a problem which is particularly serious in the case of hydrogen purification.[1]

Low-temperature Plant Heat Exchangers and Merits of Plate-fin Type

One of the earliest heat exchangers used in air separation plants is the "Hampson" coil type,[3] which consists of a bundle of small-diameter copper tubes wrapped in a special configuration round a central mandrel and contained in a cylindrical shell. Its use is limited to gas streams with a high-pressure ratio, the high-pressure stream passing through the tubes and the low-pressure stream passing across the outside of them. It is not well suited for feed gas purification as solid deposition can only take place on the outside of the tubes, which prevents them being used as reversing heat exchangers.

The most significant improvement in large air separation plant was the introduction of regenerators[3] by FRANKL. These are compact, have high thermal efficiencies, low pressure drops, are suitable for the re-evaporation of solid deposits, and are cheap. They operate on the heat-storage principle and consist of two large cylindrical vessels packed either with pancakes of wound corrugated aluminium strip or with pebbles, which store and transfer the heat between the two gas streams which flow in opposite directions through each of the vessels and are switched over periodically. Their disadvantages are, first, they are not normally suitable for simultaneous heat exchange between more than two gas streams and, second, they are not

Mr. Denton *is with the Atomic Energy Research Establishment, Harwell, and* **Dr. Ward** *is with British Oxygen Engineering Ltd., Edmonton.*

suitable for heat exchange between a high purity product and an impure feed gas because of the unavoidable contamination of the product by evaporated impurities and, third, they have varying exit gas temperatures which cause plant operating problems. They are not suitable for the product stream of a high-purity oxygen plant.

Fig. 1. A low-temperature, three-stream, plate-fin heat exchanger.

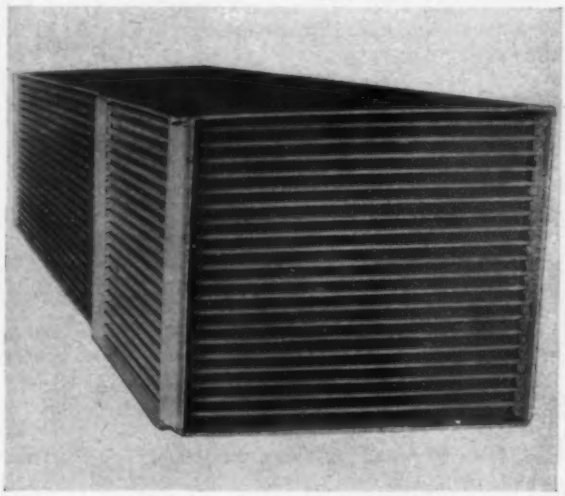


Fig. 2. A plate-fin heat exchanger block before fitting leaders.

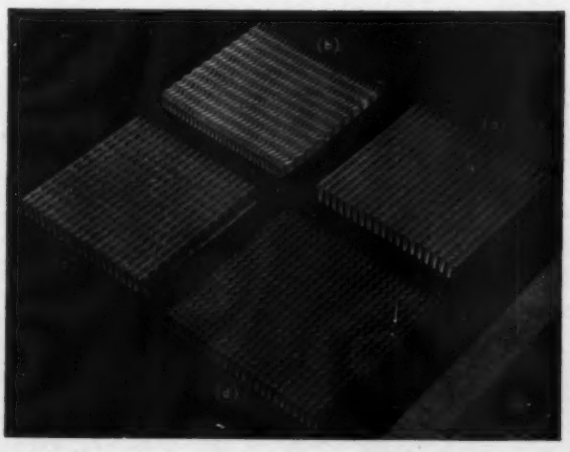


Fig. 3. Fine corrugations: (a) plain, (b) wavy, (c) louvred, (d) multi-entry.

More recently, compact tubular heat exchangers[4] have been developed for small air separation units. These consist of concentric tubes, the gas streams passing through the annular spaces which are filled with coiled copper ribbon soldered to the tube walls to provide surface extension and thermal contact. The advantages of these units are their high thermal efficiencies, low pressure drops, suitability for gas purification, and ability to handle at least three gas streams, one of which can be a high purity product. However, they are not suitable for large-scale plant because of the flow distribution problem with a large number of such tubular units in parallel.

Finally, the plate-fin heat exchanger is a development of the above type of unit and retains all its advantages, but has a rectangular rather than a circular geometry, capable of manufacture in units of sufficiently large cross-section for use in large plant. A typical three-stream plate-fin heat exchanger is shown in Fig. 1, and a view of the plate and fin construction before fitting the headers is shown in Fig. 2. It consists basically of parallel plates of metal, usually of aluminium alloy, forming a large number of parallel channels into which are sandwiched corrugated metal packings of the type shown in Fig. 3, which provide surface extension for heat transfer and mechanical strength. With suitable headers and by blocking off appropriate channels at the headers, each gas stream is made to pass through the appropriate number of channels uniformly distributed across the block section. The whole construction is brazed in a molten salt bath above 600°C. The manufacturing process is highly skilled, requiring special brazing techniques.

The pressure loading on the separating plates during operation is mainly carried in tension by the fins in the high-pressure passages, and those in the low-pressure passages provide some additional support as struts.

The four main types of fins are plain, wavy, louvred and multi-entry as shown in Figs. 3 (a), 3 (b), 3 (c) and 3 (d) respectively. Fin spacings range from 4 to 32 fins per in., depending on the type of fin and the plate spacing.

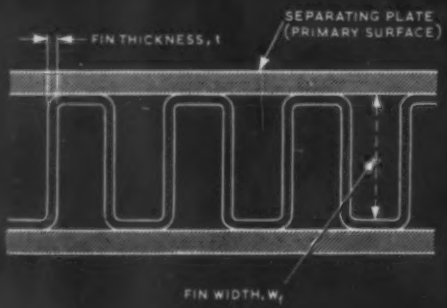
In the case of multi-entry and louvred fins, the maximum number of fins per inch is limited by the depth of their side displacement shown in Fig. 3. The maximum number of plain fins per inch manufactured at present is 32 for a plate spacing of $\frac{1}{4}$ in., which, for 0.004-in.-thick fins and 0.030-in.-thick plates, corresponds to a total heat transfer surface of 770 ft^2/ft^3, based on unit volume of the block. By comparison, a shell and tube exchanger usually has a value of only about 50 ft^2/ft^3.

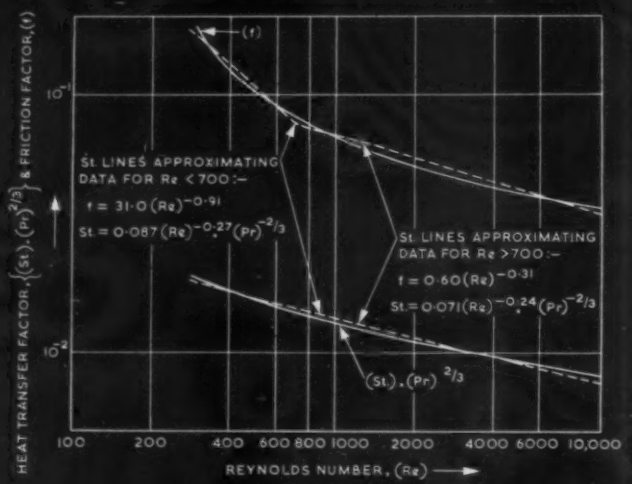
At the present time manufacturing processes limit the maximum length of a block to about 9 ft and the maximum cross-section to about $2\frac{1}{2}$ ft^2. Operating pressures are limited to about 450 psi under steady pressure conditions and about 200 psi under fluctuating pressure conditions as in a reversing heat exchanger, but these pressures exceed those used in most modern thermodynamically efficient gas separation plant.

For a standard range of blocks of relatively large volume, their cost is based primarily upon the total block volume. For large-scale plants, cost savings result from standardising block designs.

The merits of the plate-fin exchangers can be summarised as follows:

(1) High thermal efficiencies and compactness.
(2) Although not as cheap as regenerators for the case of two similar gas streams and where product purity is not important, they are the cheapest for all other low-temperature plant applications.
(3) They can provide simultaneous heat exchange between at least four gas streams in the same unit, with suitable headers. This is particularly useful in their application as reversing heat exchangers[5] where local temperature differences have often to be controlled by local recircula-





tion of one of the gas streams in an extra channel. Likewise, a high-purity product stream (e.g., oxygen) can be handled, while solid evaporation can still take place in two other reversing streams in the same unit.

(4) Plate spacings can be varied in the same unit to provide optimum heat exchanger volumes and pressure drops in gas streams of different flow rates and pressures.

(5) They have been successfully used as reversing heat exchangers in air separation plant; the finned surfaces are suited for the retention of solid deposits[2] for subsequent re-evaporation, and the geometric similarity of the channels is an advantage.

(6) Since the bulk (e.g., 75%) of the surface is very thin metal, approximately 0.008 in., they have an extremely low heat capacity; this is an important advantage for impurity solid re-evaporation when the whole unit has to be periodically warmed up and cooled down again.[1]

Design of a Counter-current Parallel-flow-Type Plate-fin Heat Exchanger

This paper only deals with the above type, as used in gas separation plant. The basic performance requirements are:

(1) The bulk temperature differences, $\triangle T_b$, between the counter-current gas streams, which depend on the relative heat capacities of the gas streams and can therefore vary along the length of the block; and (2) the pressure drops, $\triangle P$, in each gas stream in passing through the block.

Smaller temperature differences require a longer block and smaller pressure drops require a block of larger cross-section.

Calculation of Temperature Differences

To design a block it is first necessary to calculate, for each gas stream, the variation with temperature (T) of its temperature difference:

$$(\triangle t)_T = (T_b - T_s)_T \qquad \ldots . (1)$$

where T_b is the bulk gas temperature and T_s is the primary metal surface temperature.

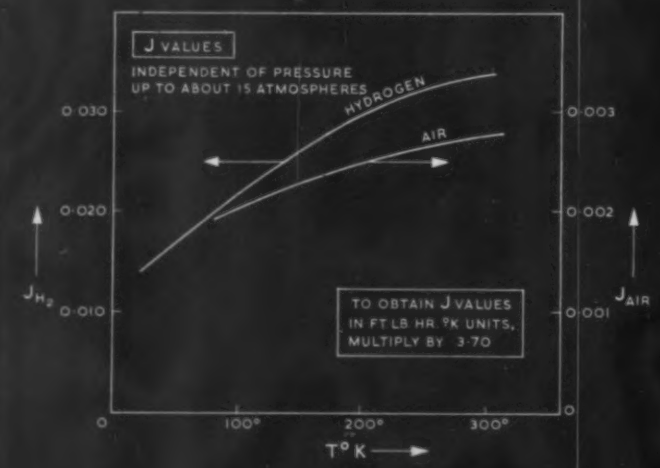
The bulk temperatures of the different gas streams at any temperature level (T) are obtained from the required bulk temperature differences ($\triangle T_b$) at specified temperature levels (usually at the ends of the block) and an overall heat balance between the gas streams, but T_s and $\triangle t$ depend on the relative heat-transfer coefficients and surface areas for the different gas streams. Thus, assuming T_s is constant* over all cross-sections, then for a two-stream exchanger with different gas streams in geometrically different channels with a bulk temperature difference $(\triangle T_b)_T$ at a temperature level (T) the corresponding temperature differences, $(\triangle t_1)_T$ and $(\triangle t_2)_T$, are given by:

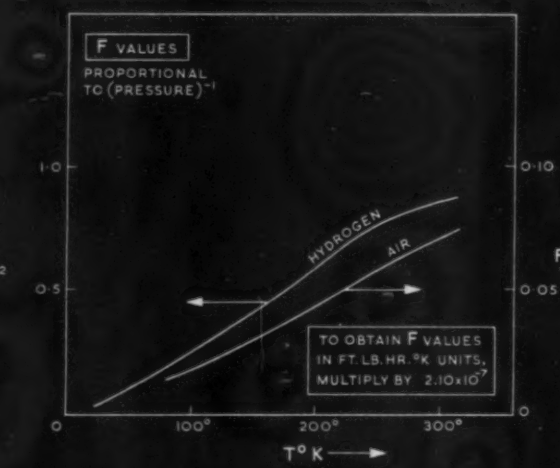
$$(\triangle t_1 + \triangle t_2)_T = (\triangle T_b)_T \qquad \ldots . (2)$$

and, from Equation (7), since the total heat transferred per unit length in each channel is the same, by:

$$(\triangle t_1)_T \,.\, [(J)_T \,.\, (S/d_e)^a \,.\, (A_1 + A_2 \,.\, \eta_f) \,.\, F_m] \quad \text{Stream 1}$$
$$= (\triangle t_2)_T \,.\, [(J)_T \,.\, (S/d_e)^a \,.\, (A_1 + A_3 \,.\, \eta_f) \,.\, F_m] \quad \text{Stream 2} \qquad \ldots . (3)$$

* Temperature drop through the metal can normally be neglected, except in extreme cases of three or more unbalanced gas streams and high heat-transfer coefficients.





where F_m is the total mass flow rate.

The cases for three or more gas streams are dealt with similarly, but are more complicated, especially if significant variations of T_s occur over the cross-section. This does not normally occur in low-temperature plant and is not dealt with in this paper.

Heat-transfer Coefficient and Pressure Drop

Experimental measurements of heat-transfer coefficient and pressure drop characteristics of typical plate-fin exchangers, covering a wide variety of fin corrugations, have been summarised by KAYS and LONDON.[6] For a given plate-fin geometry, these characteristics are expressed independently of the physical properties of the gas and of temperature and pressure by curves for the dimensionless heat-transfer factor and friction factor, respectively, plotted against REYNOLDS number (Re), using an equivalent hydraulic diameter based on total surface area per unit free volume. Fig. 5 shows characteristic heat-transfer factor—plotted as (St) . $(Pr)^{\frac{2}{3}}$—and friction factor (f) curves for multi-entry fins similar to those in Fig. 3 (d). Fig. 4 shows the relevant dimensions of a typical plate-fin channel section. The equivalent hydraulic diameter is defined by:

$$d_e = \frac{4}{(A_1 + A_2)} \qquad \ldots . (4)$$

where A_1 and A_2 are the respective primary and secondary (fins) surface areas per unit free volume; i.e., total volume less volume of metal.

Since the influence of the plate-fin geometry cannot be adequately expressed by this simple hydraulic diameter, the above curves vary for different plate-fin geometries. For plain fins there is a spread of about 20% for different plate and fin spacings.

For multi-entry and louvred fins the spread is greater still and their performance is also sensitive to the distance between the interruptions of the fin surface, e.g., their heat transfer and friction factors are increased by about 30% by reducing the distance between interruptions from $\frac{1}{4}$ in. to $\frac{3}{32}$ in.

For most multi-entry and louvred fins the heat-transfer factor is approximately double that of the corresponding plain fins, while the friction factor is somewhat more than doubled so that, for a given heat transfer and pressure-drop duty, the required block volume is about 30% less than that for plain fins. They also have the important advantage of not suffering from the characteristic instability and reduction in the heat-transfer factor in the transition region from laminar to turbulent flow, as experienced by plain fins. (See Fig. 5.)

Given the STANTON number (St) and the friction factor (f) curves for a given plate-fin channel geometry, the average surface heat-transfer coefficient (h_T) and the pressure gradient $\left(\frac{dP}{dL}\right)_T$ in the channel at a temperature, T, are given, respectively, by

$$St = \left(\frac{h_T}{C_p \,.\, G}\right) = A \,.\, Re^{-a} \, P_r^{-\frac{2}{3}} \qquad \ldots . (5)$$

and* $$f = \left(\frac{g \,.\, d_e \,.\, \rho}{2G^2}\right) \cdot \left(\frac{dP}{dL}\right)_T = B \,.\, Re^{-b} \qquad \ldots . (6)$$

where $Re = \left(\frac{G \,.\, d_e}{\mu}\right)$, $Pr = \left(\frac{C_p \,.\, \mu}{k}\right)$

and G = mass flow per unit free cross-section area. The PRANDTL number (Pr) is almost independent of temperature and pressure for a given gas and the $\frac{2}{3}$ exponent takes account of most gases which only cover a range of PRANDTL numbers from about 0.5 to 1.0, and A, B, a and b are approximately constant over reasonably wide ranges of REYNOLDS number. The constants a and b depend on the flow régime only; i.e., on the extent to which it is viscous or turbulent.

For example, in Fig. 5 for the multi-entry fins and REYNOLDS numbers between 10^3 and 10^4, $A = 0.071$, $B = 0.60$, $a = 0.24$ and $b = 0.31$.

Finally, separating the physical property dependent terms,

$$h_T = (S)^{(a-1)} \,.\, (d_e)^{-a} \,.\, (J)_T \qquad \ldots . (7)$$

and $$\left(\frac{dP}{dL}\right)_T = (S)^{(b-2)} \,.\, (d_e)^{-(b+1)} \,.\, (F)_T \qquad \ldots . (8)$$

where $$J = A \,.\, C_p \,.\, \mu^a \,./\, (\mathrm{Pr})^{\frac{2}{3}} \qquad \ldots . (7a)$$

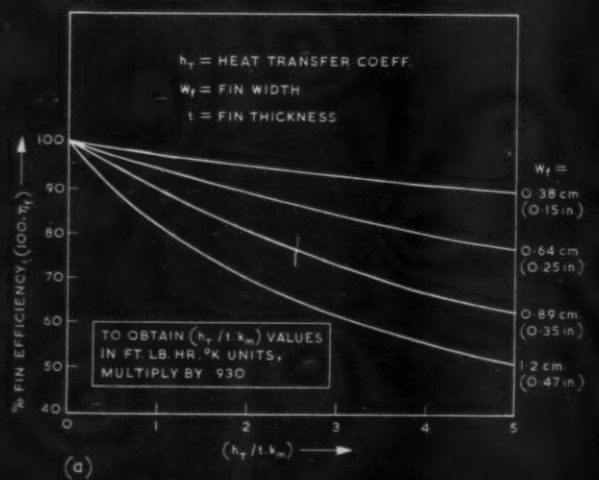
and $$F = 2B \,.\, \mu^b \,./\, g \,.\, \rho \qquad \ldots . (8a)$$

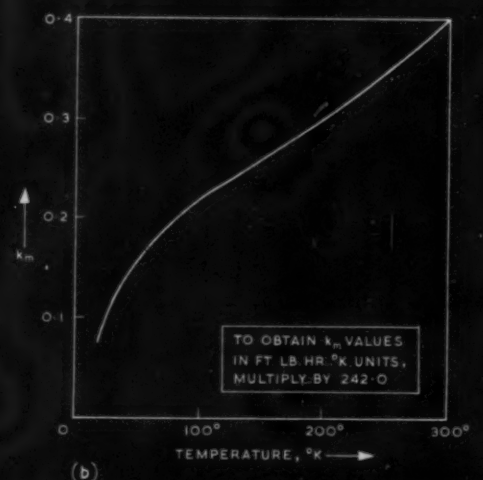
where S is the free cross-sectional area per unit mass flow rate, i.e., $S = (G)^{-1}$.

The dependence on the physical properties of the gas, on temperature and on pressure are completely taken into account by the factors J and F. They are seen to be fairly slow functions of temperature, and are plotted in Fig. 6 against T for hydrogen and air at 1 atmosphere, using the known low-temperature physical properties[9] and for the same multi-entry fins as in Fig. 5.

F is inversely proportional to pressure at all temperatures and J varies very little with pressure and only at the lowest temperatures for moderately high pressures, due to deviations from ideality.

* The gravitational constant g is included, since P is in weight units throughout.





Calculation of the Dimensions of the Heat Exchanger Block

Considering each gas stream separately, for a given variation of its temperature difference, $\triangle t$, with temperature and a given overall pressure drop, $\triangle P$, it is necesary to integrate over the full temperature range to obtain the free cross-section of block required by each gas stream and the length of the block.

If dL is the element of length required over a temperature interval T to $T + dT$, over which the pressure drop is dP, then, from Equations (7) and (8):

$$dP = \left(\frac{1}{S}\right)^{(2+a-b)} \cdot \left(\frac{1}{A_1 + A_2\,\eta_f}\right) \cdot (d_e)^{-(1+b-a)} \cdot \left(\frac{Cp \,.\, F}{\triangle t \,.\, J}\right)_T \cdot dT \quad \ldots . (9)$$

$$\text{and } dL = \left(\frac{1}{S}\right)^{a} \cdot \left(\frac{1}{A_1 + A_2\,\eta_f}\right) \cdot (d_e)^{a} \cdot \left(\frac{C_p}{\triangle t \,.\, J}\right)_T \cdot dT \quad \ldots . (10)$$

where η_f is the "fin efficiency", i.e., the overall efficiency of the fin surfaces for heat transfer, calculated in terms of the average heat-transfer coefficient (h_T), and the thermal conductivity (k_m) and thickness (t) and width (w_f) of the fins. See Figs. 7 (a) and 7 (b).

Integrating Equations (9) and (10) over the full temperature range, T_1 to T_2, the required free cross-section and length for the gas stream are then given by:

FREE CROSS-SECTION, S

$$(S)^{(2+a-b)} = \frac{(d_e)^{-(1+b-a)}}{\triangle P} \left(\frac{1}{A_1 + A_2\,\eta_f}\right) \int_{T_1}^{T_2} \left(\frac{C_p \,.\, F}{\triangle t \,.\, J}\right)_T \cdot dT \quad \ldots . (11)$$

LENGTH, L

$$L = \left(\frac{d_e}{S}\right)^{a} \left(\frac{1}{A_1 + A_2 \,.\, \eta_f}\right) \int_{T_1}^{T_2} \left(\frac{C_p}{\triangle t \,.\, J}\right)_T \cdot dT \quad \ldots . (12)$$

where the two integrals can be evaluated graphically as areas under the curves for $(C_p \,.\, F/\triangle t \,.\, J)_T$ and $(C_p \,./\triangle t \,.\, J)_T$ respectively, plotted against T between T_1 and T_2. Any significant variation of pressure along the gas stream need only to be taken account of in evaluating F, which is inversely proportional to pressure.

An example is calculated in the Appendix.

Dependence of Dimensions of Heat Exchanger Block on Fin and Plate Spacings

Without introducing the physical properties of the gas, from Equations (5) and (6):

$$dL = \left(\frac{1}{\mathrm{St} \,.\, \triangle t}\right)_T \cdot \left(\frac{1}{A_1 + A_2 \,.\, \eta_f}\right) \cdot dT \quad \ldots . (13)$$

$$\text{and } dP = \left(\frac{f/\mathrm{St}}{\triangle t}\right)_T \cdot \left(\frac{1}{2\rho g\, S^2}\right) \left(\frac{A_1 + A_2}{A_1 + A_2 \,.\, \eta_f}\right) \cdot dT \quad \ldots . (14)$$

These equations show:

(a) that since $\eta f \simeq 1$ and $(f/St) \simeq$ a constant which is almost independent of plate and fin *spacings* and REYNOLDS number,[6] the required cross-section is independent of the fin geometry and is determined only by the overall pressure drop and temperature differences, except that the constant (f/St) varies for different *types* of fins[6], i.e., plain, multi-entry, etc., generally being higher for those with enhanced values of St; and

(b) that the required length, and hence the volume, decreases faster than in proportion to d_e as the plate and fin spacings are reduced.

This saving is only offset, for very close spacings, by an increase in (f/St) due to increased inertia losses, which then requires an increase in cross-section to restore the required pressure drop. Before this limitation on spacings is reached, the volume is seen to be reduced as $(d_e)^{1.3}$ for high REYNOLDS numbers ($a \simeq 0.3$) and more rapidly still at low REYNOLDS numbers, approaching $(d_e)^2$ for ideal viscous flow, when flow conditions approach those in a highly efficient packed regenerator.

However, these low REYNOLDS numbers do not occur for the high densities and low viscosities in low-temperature plant, even for relatively close fin spacings and a light gas, as shown in Table I of a later paper. Also practical limitations to reducing spacings are:

(1) manufacturing limitations;
(2) volume of metal itself becomes too high; and
(3) longitudinal heat conduction through the metal block and gas distribution problems arise as the cross-section to length ratio increases for low REYNOLDS numbers.

Saving in Heat Exchanger Volume by Varying the Cross-section of the Block

Over large temperature ranges there is, in principle, a case for a saving in total block volume by reducing the cross-section in one or two stages in passing from room temperature to the lowest temperature. At the lower temperatures, mainly because of the rapid reduction in gas density, the pressure drop decreases rapidly while the heat-transfer coefficient does not do so, so that the cross-section can be reduced accordingly. For a given overall heat transfer and pressure-drop duty, considering the ideal case of the cross-section S varying continuously along the heat exchanger, its optimum value at a temperature, T, for a minimum total volume is when the ratio of volume to pressure drop has the same optimum value everywhere along the exchanger, i.e., when $\left(S \,.\, \frac{dL}{dP}\right)_T$ in Equations (9) and (10) is kept constant or when:

$$(S_T)_{opt.} \propto (F_T)^{\frac{1}{(3-b)}} \quad \ldots . (15)$$

independently of overall pressure drop and temperature differences. The constant of proportionality depends on the overall $\triangle P$ and the $\triangle t$ distribution and is obtained by integrating Equation (9) with $(S_T) = (S_T)_{opt.}$ Finally, by integrating Equation (10) with $(S_T) = (S_T)_{opt.}$, the maximum ideal volume reduction factor is given by:

$$\frac{\left\{\int_{T_1}^{T_2} \left(\frac{C_p \,.\, (F)^{\left(\frac{1-a}{3-b}\right)}}{\triangle t \,.\, J}\right)_T dT\right\}^{\left(\frac{3-b}{2+a-b}\right)}}{\left\{\int_{T_1}^{T_2} \left(\frac{C_p}{\triangle t \,.\, J}\right)_T dT\right\} \left\{\int_{T_1}^{T_2} \left(\frac{C_p \,.\, F}{\triangle t \,.\, J}\right)_T dT\right\}^{\left(\frac{1-a}{2+a-b}\right)}} \quad \ldots . (16)$$

and depends only on the $\triangle t$ distribution and the temperature range. For a counter-current unit (multi-entry fins) over a large temperature range between 300°K and 20°K, for hydrogen, $\left(\frac{S_{300}}{S_{20}}\right)_{opt.} = 3.7$, and, for two extreme cases of the ratio of warm-end to cold-end temperature differences of 5 and $\frac{1}{5}$, the maximum ideal volume reduction factors are only 0.85 and 0.90 respectively.

Thus, despite the large variation in the optimum cross-section, even the maximum ideal volume saving is small, so it is not worth varying the cross-section in practice, except for very low temperature processes.

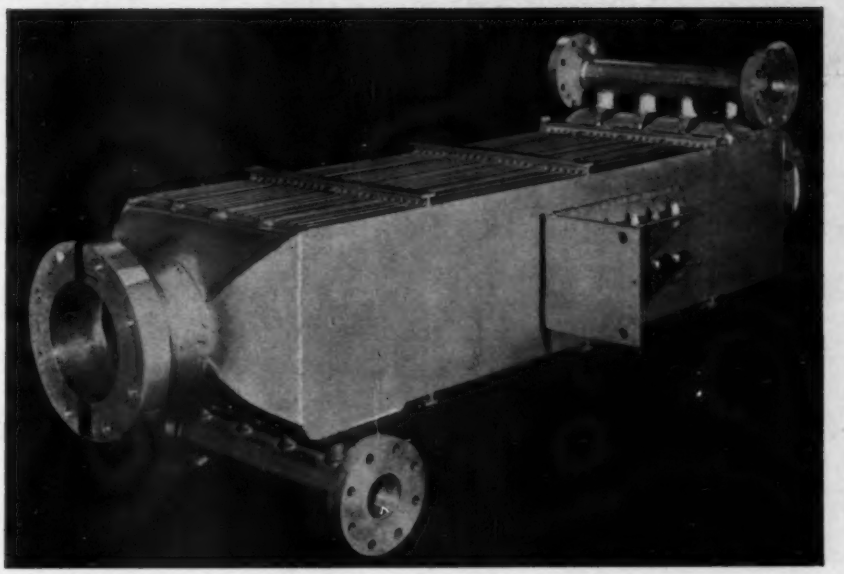


Fig. 8. A low-temperature plate-fin heat exchanger for a high-pressure and a low-pressure steam with small tanks and manifolding for the high-pressure headers

Header Design, Pressure Losses and Gas Distribution

The previous calculations provide the block pressure drop only. To this must be added the header pressure losses and, where appropriate, the internal distributor pressure losses. The design of the headers is important not only because of the additional pressure drop they impose, but also because of their possible effect on gas distribution in the heat exchanger. To ensure good gas distribution, pressure losses in the headers and manifolds should be kept as small as possible in relation to the overall block pressure drop. When designing a low-pressure drop header there is a limit to the maximum size of the header "tank" due to the pressure stressing of the tank material. Thus, for high pressures, several smaller tanks have to be used in parallel with a manifold, e.g., for a test pressure of about 24 atm. individual tanks of 6 in. × 21 in. cross-section are used in parallel. In certain high-pressure applications, to achieve good gas distribution, even the block cross-section may be controlled by header design and overall pressure losses are unimportant. Fig. 8 shows a two-stream unit for a high-pressure stream and a low-pressure stream with small tanks and manifolding for the high-pressure headers.

The header pressure losses are expressed as an equivalent number of velocity heads, N, corresponding to a pressure drop of $N \cdot \frac{\rho v^2}{2g}$ where v is a gas velocity. For a simple single-block layout the principal entrance header losses are due to the expansion from the entrance pipe to the entrance header tank, $\triangle P_1$, and the contraction from the tank into the block, $\triangle P_2$. One velocity head, based on the velocity in the pipe, should be allowed for $\triangle P_1$, and $\triangle P_2$ can be determined from Fig. 9 in which the number of velocity heads, based on the velocity in the block channels, is plotted[7] as a function of R_A, the ratio of the cross-sectional area of the tank to the corresponding free cross-sectional area of the block.

Similarly, the principal exit header losses are due to the expansion from the block to the exit header tank, $\triangle P_3$ and the contraction from the tank to the exit pipe, $\triangle P_4$. The expansion loss, $\triangle P_3$, can be determined from Fig. 9, and half a velocity head, based on the velocity in the pipe, should be allowed for $\triangle P_4$. Further pressure losses have to be allowed for manifolds with multiple header tanks and when several blocks are in parallel.

Inefficiencies Due to Maldistribution of Gas Flow

When a heat exchanger is divided into sections arranged in parallel and not in thermal contact with each other, a serious reduction in overall efficiency results if the distribution of the gas flows between the sections is not correct. This effect is most serious for high ratios of temperature span to temperature difference.

Such flow maldistribution can occur in plate-fin units both within the blocks themselves and between blocks when they are used in parallel. Adequate distribution within the blocks themselves is obtained by correct design and ensuring that the block pressure drop is greater than that in the headers, while the high conductivity of the metal helps to compensate for any residual flow maldistribution. Maldistribution between blocks in parallel is a much more serious problem. First, the blocks have to be flow tested to establish their pressure-drop characteristics and, where necessary, selective assembly carried out to match the characteristics of each parallel set of units. Secondly, the manifold systems have to be designed to give equal estimated losses in the pipework associated with each block and to give additional pressure drops which are small compared to those in the blocks themselves.

To illustrate the seriousness of maldistribution effects, the case of maldistribution between two blocks in parallel will be considered for two counter-current gas streams with a given mass flow ratio (r) and temperature span ($\triangle T$). Fig. 10 shows the conditions in one of these blocks. Assuming that the specific heat, C_p, of the two gas streams and the overall surface heat-transfer coefficient, h, between them does not vary with temperature, an overall log mean temperature difference, δT_m, will take account of the variation in temperature differences when $r \neq 1$. Thus the heat transferred per unit mass flow of stream Y is:

$$(\text{Area}) \times h \times \delta T_m = C_p \cdot \triangle T_Y$$

where h can be regarded as a constant, since it is not sensitive to the small changes in flow ratio considered here. Thus, for a given heat exchanger block and nominal mass flow rate there is a characteristic constant

$$K = \frac{\triangle T_Y}{\delta T_m} \qquad \ldots . (17)$$

which is independent of r and can be considered as the "number of heat-transfer units", where:

$$\delta T_m = \frac{(\delta T_w - \delta T_c)}{ln\,(\delta T_w/\delta T_c)} \quad \ldots . (18)$$

From heat balances:

$$\delta T_w = \triangle T - \triangle T_Y$$

and

$$\delta T_c = \triangle T - \frac{\triangle T_Y}{r}$$

Solving for δT_c, which is proportional to the amount of heat allowed to pass to the lower temperature (i.e., the refrigeration load imposed at the lower temperature),

$$\delta T_c = \frac{(1-r)\,.\,\triangle T}{(e^{\frac{K(1-r)}{r}} - r)} \quad \ldots . (19a)$$

and when $r = 1$,

$$\delta T_c = \triangle T/(1+K) \quad \ldots . (19b)$$

If the flow maldistribution between two such identical blocks in parallel is such that, instead of the flow ratio (r) being the same in both blocks, it has the values ($r + \triangle r$) and ($r - \triangle r$), respectively, the value of the resulting cold-end temperature difference, δT^1_c, between the two gas streams when they are recombined is given by:

$$\delta T_c{}^1 = \frac{1}{2r}\left\{(r+\triangle r)(\delta T_c)_{r+\Delta r} + (r-\triangle r)(\delta T_c)_{r-\Delta r}\right\} \ldots . (20)$$

where $(\delta T_c)_{r\,+\,\Delta r}$ and $(\delta T_c)_{r\,-\,\Delta r}$ are given by Equation (19).

Thus for a heat exchanger block whose constant, K and flow ratio, r, are given, the relative increase in the cold-end temperature difference due to a flow maldistribution of $\triangle r$ between two such blocks is given by Equations (19) and (20).

Values of $(\delta T^1_c/\delta T_c)$ are plotted in Fig. 11 for the case $r = 1$, for $K = 10$, 50 and 100 and for $\triangle r$ up to 0.10, i.e., 10% flow maldistribution.

This shows how the effect becomes rapidly more serious as K is increased, i.e., as the ratio of temperature span to temperature difference is increased. For example, with $K = 50$, as occurs in air separation plant (i.e., to cool from 290°C to 80°K with a temperature difference of 4°) only a 5% flow maldistribution would cause a 50% increase in δT_c and a corresponding increase in the refrigeration load which the heat exchanger is designed to reduce. For $K = 100$ the refrigeration load would increase by 160% for 5% maldistribution.

Plate-fin Exchangers as Reversing or Dual Alternating Units for Low-temperature Gas Purification

The heat exchangers which cool the impure feed gas to a gas separation plant will rapidly block if the solid impurity deposits are not continuously re-evaporated. The simplest method is to operate them as reversing exchangers when, by periodically switching over the two counter-current gas stream channels, the solid deposits are re-evaporated into the lower-pressure gas stream leaving the plant. The plate-fin unit is particularly suitable for this operation because the two channels can be made geometrically the same and a suitable type of fin corrugation can handle the solid deposition.

A general theory of the operation of a reversing heat exchanger has been described previously[1] and verified experimentally.[2] The main design requirement is that to achieve complete evaporation of solid the temperature differences between the two gas streams must be everywhere below certain limiting maximum values which can be calculated[1] from the solid-vapour phase equilibria for the impurity concerned. If the process cycle does not provide adequately low temperature differences in the region of deposition, these can be readily provided by the local

recirculation[5] of one of the gas streams in a third channel in the block.

Recently reported[2] experimental results describe the very satisfactory performance of a plate-fin reversing exchanger unit, using multi-entry type fins, handling up to 1% of carbon dioxide impurity in hydrogen in the temperature range 290°K to 80°K. This gas mixture is an extreme example of one in which the impurity becomes super-saturated (the heat-transfer rate is greater than the impurity diffusion rate) so that a large fraction of the impurity tends to condense as solid particles in the ingoing gas stream and be transported to lower temperature levels when it would not be evaporated. However, in the above experiments the multi-entry type fins provided efficient solid nucleation centres and retained all the solid, thus overcoming this problem.

Because of the good retention of the solid deposits, the plate-fin unit is also suitable for use as a dual alternating (or blocking) purification exchanger, when a pair of exchangers are alternated, one on stream condensing out the solid impurity while the other is off stream being warmed up and all the solid being evaporated from it before being cooled down and returned on stream again. This system[1, 8] is not as elegant as the reversing exchanger, but would be used, for example, where the solid-vapour phase equilibria or the operating temperature differences do not meet the above reversing exchanger requirements. For this application the heat capacity of the plate-fin unit is very low, since 75% of its surface area consists of very thin fin corrugations. This is a definite advantage, as it reduces to a minimum the refrigeration load associated with warming and cooling the units. This advantage is exploited in a scheme proposed[1] for hydrogen purification in the range 50°K to 20°K.

REFERENCES

[1] Denton, W. H., Shaw, B., and Ward, D. E. "Purification of Hydrogen for Distillation, Part I", *Trans. Inst. Chem. Engrs.*, 1958, **36**, No. 3, 179.
[2] Denton, W. H., Shaw, B., Gayler, R., and Seager, P. "Purification of Hydrogen for Distillation, Part II", Trans. Inst. Chem. Engrs., read on March 17 1959, at the Spring Meeting.
[3] Ruhemann, M. "Separation of Gases", Oxford University Press, 1940.
[4] Trumpler, P. R., and Dodge, B. F. *Trans. Am. Inst. Chem. Engrs.*, 1947, **43**, No. 2, 75.
[5] Lobo, W. E., and Skaperdas, G. T. *Trans. Am. Inst. Chem. Engrs.*, 1947, **43**, No. 2, 69.
[6] Kays, W. M., and London, A. L. "Compact Heat Exchangers", The National Press, California, U.S.A., 1955.
[7] Kays, W. M. "Loss Coefficients for Abrupt Changes in Flow Cross-section", *Trans. Am. Soc. Mech. Engrs.*, 1950, 1067.
[8] Crawford, D. B. *Chem. Eng. Prog.*, February, 1950. 74.
[9] "Tables of Thermal Properties of Gases", National Bureau of Standards, Circular No 564, U.S. Dept. of Commerce.
[10] Denton, W. H. "Plate-fin Heat Exchangers—Overall Economics", see p. 40 this issue.

ACKNOWLEDGMENTS

Thanks are due to Marston Excelsior Ltd. of I.C.I., manufacturers of plate-fin heat exchangers, and in particular to Mrs. A. G. Lenfestey for many helpful discussions and their permission to publish Figs. 1, 2, 3, 5 and 8.

SYMBOLS USED

For definitions of the symbols used in this article, readers are referred to "Plate-fin Heat Exchangers—Overall Economics", p. 40 of this issue.

Appendix

Example of a Calculation of the Block Dimensions of a Low-temperature Plate-fin Heat Exchanger

Units of ft lb hr °K will be used throughout this Appendix. The appropriate conversion factors are given in Figs. 6, 7(a) and 7(b).

A reversing plate-fin heat exchanger is required to cool 1000 lb/hr of hydrogen at 5 atm. from 150°K to 50°K by 1037 lb/hr of hydrogen at 1.5 atm. and 49°K. The permissible pressure drop in the low-pressure stream is 0.04 atm. and a 20% design margin on heat-transfer performance is required.

Plate-fin Geometry

Since it is a reversing exchanger, the high- and low-pressure channels will have the same cross-section and plate-fin geometry. (Otherwise the high-pressure channels could have a smaller cross-section.) Multi-entry fins will be chosen and, to give a small block volume, relatively close plate and fin spacings will be chosen with the following dimensions:

Plate spacing	= 0.250 in. = 0.0208 ft
Plate thickness	= 0.030 in. = 0.0025 ft
Number of fins per inch	= 12
Fin thickness	= 0.006 in. = 0.0005 ft
∴ Fin spacing	= 0.077 in. = 0.00645 ft

Thus:

$$\text{Primary surface area, } A_1 = \frac{2}{0.0208 - 0.0005} = 98 \text{ ft}^2/\text{ft}^3$$

$$\text{Secondary surface area, } A_2 = \frac{2}{0.00645} = 310 \text{ ft}^2/\text{ft}^3$$

$$\text{Equivalent hydraulic diameter, } d_e = \frac{4}{(98 + 310)} = 0.010 \text{ ft}$$

$$\text{and } t = 0.0005 \text{ ft}$$

Calculation of J and F Values

The heat-transfer and friction factor relationships for these multi-entry fins are plotted in Fig. 5 and can be expressed as follows:

$\text{Re} > 700$

$$\text{St} = 0.071\ \text{Re}^{-0.24}\ \text{Pr}^{-2/3}$$
$$\text{and } f = 0.60\ \text{Re}^{-0.21}$$
$$\text{giving } a = 0.24 \text{ and } b = 0.31$$
$$\text{and } A = 0.071 \text{ and } B = 0.60$$

$\text{Re} < 700$

$$\text{St} = 0.087\ \text{Re}^{-0.27}\ \text{Pr}^{-2/3}$$
$$\text{and } f = 30.4\ \text{Re}^{-0.91}$$
$$\text{giving } a = 0.27 \text{ and } b = 0.91$$
$$\text{and } A = 0.087 \text{ and } B = 30.4$$

The corresponding values of J and F are then obtained from Equations (7a) and (8a); they are plotted in Fig. 6 for $\text{Re} > 700$.

Temperature Difference Distribution

Since the two channels are geometrically the same, the temperature difference, $\triangle t$, is the same in both gas streams and equal to half the overall temperature difference between the two streams.

From a heat balance, using data[9] for the total heat of hydrogen and assuming no heat leak, the warm-end overall temperature difference is 3°K, the cold-end overall temperature difference is 1°K, and the distribution of the temperature difference, $\triangle t$, in each gas stream is as shown in Fig. 12.

Values of $(C_p/\triangle t \,.\, J)_T$ and $(C_p \,.\, F/\triangle t \,.\, J)_T$ are then plotted over the full temperature range (see Fig. 12), using the appropriate J and F values plotted in Fig. 6 for hydrogen and the multi-entry fins, assuming for the present that $\text{Re} > 700$. The pressure dependent quantity F is based on 1.5 atm. for the low-pressure stream.

Pressure Drop

The pressure drop is greatest in the low-pressure stream and the permissible value is 0.04 atm. Therefore, the design is based on the low-pressure stream conditions for a pressure drop of $\frac{0.04}{1.2} = 0.033$ atm.

$$\text{i.e.: } \triangle P = 70.0 \text{ lb./ft}^2$$

after allowing for a 20% increase in the calculated length of the exchanger for the design margin on heat transfer performance.

Calculation of the Free Cross-section, S, of Block

From Equation (11), using the values for a and b for $\text{Re} > 700$, the required free cross-section of the heat exchanger (based on the low-pressure stream conditions) is given by:

$$S^{1.93} = \frac{1}{\triangle P} \cdot \frac{(d_e)^{-1.07}}{(A_1 + A_2\,\eta_f)} \int_{49^\circ}^{147^\circ} \left(\frac{C_p \,.\, F}{\triangle t \,.\, J}\right)_T . \, dT$$

where $\triangle P = 70$;
$d_e = 0.01$;
$A_1 = 98$;
and $A_2 = 310$.

We will assume, for the present, that $\eta_f = 0.85$. The integral is equal to 1.28×10^{-4}, from Fig. 12.

$$\therefore S = 6.3 \times 10^{-4} \text{ ft}^2 \text{ lb.}^{-1} \text{ hr.}$$

REYNOLDS *Number*

The REYNOLDS numbers must now be calculated to see if they lie within the range assumed for the heat transfer and friction factor relationships, i.e., $10{,}000 > \text{Re} > 700$ in this case.

Thus:

$$Re_{147^\circ} = \frac{d_e}{S \,.\, \mu_{147^\circ}} = 1200$$
$$\text{and } Re_{49^\circ} = 2700$$

so that the REYNOLDS numbers are well within the assumed range.

Fin Efficiency

Also, the fin efficiency must be calculated to see if it is at least not less than the assumed value, i.e., $\eta_f = 0.85$ in this case.

From Equation (7), using the same a and b values,

$$h_{49^\circ} = (S)^{-0.76} \cdot (d_e)^{-0.24} \cdot J_{49^\circ}$$
$$= 54.0$$

and, from Fig. 7 (b),

$$k_m = 37.0 \text{ at } 49^\circ\text{K.}$$

$$\therefore \left(\frac{h_T}{t \cdot k_m}\right)_{49^\circ} = 2900$$

when, from Fig. 7 (a),

$$\underline{\eta_f = 0.84 \text{ at } 49^\circ\text{K.}}$$

and, similarly,

$$\underline{\eta_f = 0.85 \text{ at } 147^\circ\text{K.}}$$

Therefore, the assumed value of 0.85 is only just low enough. If this calculated fin efficiency is very different from the assumed value, the cross-section should be recalculated using a safer value.

Calculation of Length, L, of Block

From Equation (12), using the same a and b values, the required length is given by:

$$L = \left(\frac{d_e}{S}\right)^{0.24} \cdot \left(\frac{1}{A_1 + A_2 \cdot \eta_f}\right) \cdot \int_{49^\circ}^{147^\circ} \left(\frac{C_p}{\triangle t \cdot J}\right)_T \cdot dT$$

where $d_e = 0.01$;
$S = 6.3 \times 10^{-4}$;
$A_1 = 98$;
$A_2 = 310$; and
$\eta_f = 0.85$;

and the integral is equal to 4.63×10^3, from Fig. 12.

$$\therefore \underline{L = 25.0 \text{ ft.}}$$

Final Block Dimensions

Allowing a 20% design margin on heat-transfer performance, the actual length of the block will be 30 ft.

To allow for the cross-section of the block occupied by the metal, the free cross-section must be multiplied by the factor:

$$\left(\frac{0.250 + 0.030}{0.250 - 0.006}\right) \cdot \left(\frac{0.077 + 0.006}{0.077}\right) = 1.24$$

Therefore, the actual cross-section of the block will be:

$$1.24 \times 2 \times 1037 \times 6.3 \times 10^{-4} = \underline{1.62 \text{ ft}^2.}$$

This takes account of both gas streams and is based on the flow of the low-pressure gas stream.

If we assume, for example, that the maximum length of a block is limited to 9 ft and the maximum cross-section to 2.5ft², this exchanger would consist of a set of four standardised blocks connected in series, each 7 ft 6 in. long and with a square cross-section of 15.2 in. × 15.2 in. excluding the thicker outside containing plates. Each block would have 54 gas passages (27 per gas stream), each $\frac{1}{4}$ in. wide, i.e., 55 plates, including the two outside containing plates.

For the same heat exchanger specification, but larger flows, a set of four such blocks in series would be used up to their maximum cross-section of 2.5 ft² to handle up to 1.5 times the above flows. For still larger flows, two duplicate sets, each of four such standardised blocks and with the same cross-section, would be used in parallel to handle up to three times the above flows, and so on.

N.B. In the above example, $\triangle t$, C_p, J and F vary considerably with temperature and there is an unusual distribution of $\triangle t$, so that the integrals in Equations (11) and (12) have to be evaluated to obtain sufficiently accurate results. However, in several cases, particularly for small temperature ranges, the log mean temperature difference and average values of C_p, J and F, at the average temperature, can be used instead of evaluating the above integrals.

Water Treatment Plant for Fords

WHEN considering plans for the new paint, trim and assembly building at Ford's Motor Works, Dagenham, a double problem concerning water supply confronted the management. Both problems affected vehicle body production. The first was the possibility of a restriction in the supply of mains water coupled with a recent increase in water charges. No alternative source of supply was available and Ford's production engineers were worried, since it was envisaged that about 25,000 gal./hr or 400,000 gal. a day would be required. The second problem was that mains water contains dissolved solids, the presence of which could affect the finishing processes.

The Permutit Co. Ltd., London, were asked to investigate the question of water treatment to overcome these difficulties economically. The scheme proposed by Permutit and applied in Ford's comprehensive plant recovers the waste water from the Paint Spray Washmobile Section and Wet Sand Deck, purifies it and recirculates it for re-use; at the same time it produces demineralised water for the Washmobile. Normal output is 23,000 gal./hr of purified, reclaimed waste water and 6300 gal./hr of demineralised water.

Waste water reclamation involves a process of chemical treatment, flocculation, sedimentation and filtration which removes suspended solids from the recovered water and destroys harmful bacteria. The waste water from the Paint Spray Washmobile and Wet Sand Deck is collected by a drain system and delivered to a tank from which it flows over a weir and dosed with chemicals. The chemicals used are lime suspension and a mixture of chlorine and ferrous sulphate. These chemicals sterilise the contaminated water and interact to form ferric hydroxide—a floc forming substance which coagulates suspended solids. Coagulation is assisted by two paddle wheels which agitate the flocculated water as it flows through a channel into sedimentation tanks. Here the heavier coagulated solids settle as sludge, which is removed periodically from the bottom of the tanks and discharged to drain. Partially clarified and settled water collects at the top and flows to rapid gravity sand filters where residual suspended solids are removed. The filtrate is then recirculated for re-use by the Wet Sand Deck, a proportion being by-passed through the "Deminrolit" plant *en route* to the Washmobile.

The by-passed filtrate is blended with mains water in a break pressure tank. The mains water is introduced at this point for convenience and to make up losses resulting from periodical sludge removal and filter washing in the waste reclamation point. The blended water flows through a two-stage Deminrolit plant comprising a pair of hydrogen-ion cation exchange units and a pair of De-acidite anion exchange units. These units are arranged as pairs in parallel and the water flows first through the H.I. units and then through the De-acidite units. In the H.I. units dissolved salts are converted to their corresponding acids, which are then absorbed as the water passes through the De-acidite units. CO_2 is liberated by the breakdown of the dissolved bicarbonate salts in the H.I. units. This is removed in a degassing tower where the water falls through an upstream of blown air which picks up the CO_2 and vents it to atmosphere. The de-gassed water collects in a tank at the base of the tower where it is dosed with a small quantity of dilute caustic soda solution. This neutralises residual CO_2 and maintains correct *p*H conditions. The water is then pumped to the Washmobile via a reserve storage tank.

The benefits afforded by this plant are considerable. The total consumption of mains water required by the new body-finishing building is reduced from 400,000 gal. daily to 32,000 gal. daily, (thereby cutting water charges) and obviating the risk of an interruption in the supply.

This type of plant has much to offer to manufacturers in other industrial fields, including the chemical industry.

SECONDARY SURFACE HEAT EXCHANGERS

The construction, use and heat-transfer characteristics of the latest type of compact low-temperature exchanger

by A. G. LENFESTEY, B.A.

THE use of secondary surface plate-and-corrugation heat exchangers has long been established for aircraft applications for which their compact form and low weight are ideally suited. During the 1930s this type of construction was introduced by Marston Excelsior Ltd. for aero-engine radiators, using dip-soldered copper. Since that time this basic form of construction has been widely used and developed for an ever-increasing range of applications and materials. Process developments have led on the one hand to inert atmosphere furnace-brazing of copper, nickel, steel, titanium and various non-ferrous alloys, and, on the other hand, to flux dip-brazing of aluminium alloys, the latter process being operated under licence from A.L.C.O.A.

The combination of the plate-and-corrugation construction and the dip-brazing process has resulted in an extremely adaptable design basis for high-efficiency, compact heat exchangers. These are particularly well suited to low-temperature applications, not only for their performance and compactness, but also for their low heat capacity and for the excellent ductility and increasing strength of aluminium alloys at sub-atmospheric temperatures. The manufacturing techniques and equipment have been developed progressively to produce individual elements of considerably greater size than are at present made in furnace-brazed constructions, and the argon-arc welding process has contributed materially in the attachment of header tanks and pipework and in final assembly work.

During recent years a wide variety of dip-brazed aluminium alloy plate-and-corrugation heat exchangers has been designed, manufactured and successfully operated in low-temperature plants, both on an experimental and on a "tonnage" scale. Undoubtedly the use of such units will become even more widespread in the future.

Mrs. Lenfestey *is with Marston Excelsior Ltd., Wolverhampton (a subsidiary company of I.C.I. Ltd.).*

Method of Construction and Mechanical Design

The basic method of construction is extremely flexible, and can be efficiently applied to many differing conditions of operation. Fig. 1 shows the method of assembly of a single element or passage, consisting of two flat plates, one strip of corrugation and two edge-sealing channels. This method is used for furnace-brazed constructions and for dip-brazed aluminium alloys. The brazing medium is incorporated in the build-up to provide uniform bonds throughout the whole assembly. Most careful control is exercised to ensure the quality and uniformity of these bonds which must provide both the thermal contact between the flat "primary surface" plates and the "secondary surface" corrugations, and also the mechanical ties to withstand internal pressure loadings.

Flow Patterns

Fig. 2 illustrates the two basic methods by which a number of elements may be built up to provide either a

TABLE 1. Geometrical Data for Typical Corrugations
(This does *not* represent the complete range)

Type	Height w_f in.	Fins per in.	Thickness t in.	a ft²	A_1 ft²	A_2 ft²	d ft
Plain	0.25	10	0.010	0.0180	1.80	4.80	0.0109
,,	0.35	12	0.010	0.0249	1.76	8.15	0.0100
,,	0.464	12	0.010	0.0334	1.76	10.90	0.0106
Multi-entry	0.15	12	0.010	0.0102	1.76	3.36	0.0080
,,	0.25	12	0.012	0.0170	1.71	5.71	0.0092
,,	0.35	15	0.008	0.0251	1.76	10.26	0.0083
,,	0.464	15	0.008	0.0335	1.76	13.70	0.0086
Herringbone	0.15	18	0.010	0.0096	1.64	5.04	0.0057
,,	0.35	18	0.010	0.0232	1.64	12.23	0.0067

a = free-flow area per foot width of corrugation; A_1 = primary surface area per ft width × ft length of corrugation; A_2 = secondary surface area per ft width × ft length of corrugation; d = hydraulic mean diameter = $\frac{4a}{\text{wetted perimeter}} = \frac{4a}{(A_1 + A_2)}$ for areas specified above.

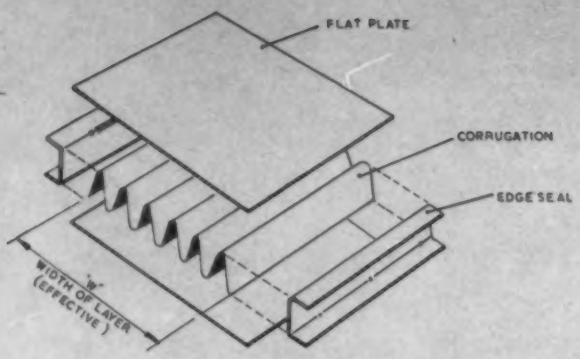


Fig. 1. Method of assembly of element.

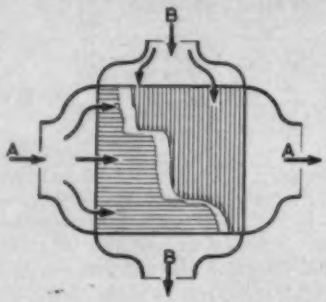


Fig. 2. Typical arrangement of elements.

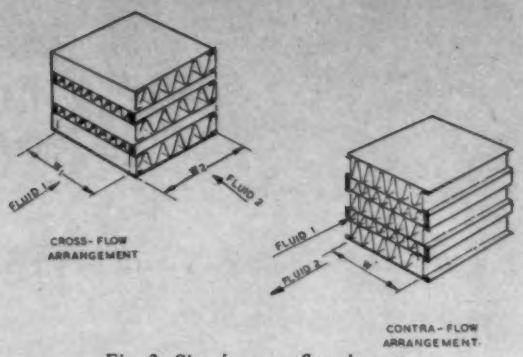


Fig. 3. Simple cross-flow layout.

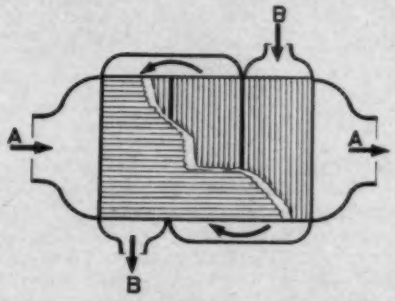


Fig. 4. Cross-contra-flow layout.

cross-flow or a contra-flow unit. Either of these arrangements may be further developed to produce multi-pass or multi-stream configurations by including suitable internal seals and distributors and external header tanks. Some typical arrangements are shown in Figs. 3-7.

The simple cross-flow layout in Fig. 3 is generally suitable for low to moderate duties. For higher duties, when the mean effective temperature difference in cross-flow is significantly below that in contra-flow, the use of a cross-contra-flow pattern, as illustrated in Fig. 4, frequently offers an efficient and compact solution. This arrangement is used extensively both for gas/gas and for gas/liquid applications, especially when handling a low-pressure gas stream on one side of the heat exchanger.

For many low-temperature applications a very close temperature approach relative to the overall temperature range is required, and in such cases contra-flow units are generally provided. Headering arrangements must be matched to the type of duty involved. Fig. 5 shows an arrangement which is mainly suitable for low-pressure applications, while Fig. 6 illustrates a symmetrical arrangement suitable for reversing duty and for large-size high-pressure units. For more than three streams an arrangement such as Fig. 7 may be used.

Corrugations

With any of these flow arrangements, the corrugations may also be varied. A comprehensive range has been developed to cater for widely differing requirements, and includes plain, straight-through passages, pierced "multi-entry" passages and continuously waved "herringbone" passages. Typical examples are specified on Table 1. For standard industrial applications, heights vary between 0.15 in. and 0.464 in., thicknesses between 0.008 in. and 0.015 in. and fin pitching from 10 to 15 or 18 fins per in. Surface areas range from about 300 to 450 sq. ft per cu. ft block volume, and free-flow areas from about 0.75 to 0.80 sq. ft per sq. ft cross-section. Each of these may be apportioned between the various streams to suit requirements. In general, the 0.15 in. or 0.25 in. high corrugations are used for liquid or condensing streams, and the taller corrugations for gas streams, though there are exceptions. The "multi-entry" and "herringbone" patterns give enhanced heat-transfer characteristics and are particularly attractive for applications involving close temperature approaches (i.e., characteristically long units).

Manufacture

Each unit is assembled directly into its brazing jig. After brazing, the unit is removed from the jig, thoroughly cleaned and leak tested. Header tanks are then fitted, generally by Argon-arc welding. Fig. 8 shows a cross-contra-flow unit with the high-pressure multi-pass tanks in position, and with detachable tanks on the low-pressure straight-through stream. This illustrates the relationship between the internal block geometry and the header tanks. For industrial applications, the end tanks are normally butt-welded in position. A set of small contra-flow units is shown in Fig. 9. The left-hand unit is a simple two-stream unit, while those on the right each carry a third stream. These units are typical of many which have been supplied for small and medium-sized plants.

Composite Assemblies

On tonnage-scale plants it is frequently necessary to build up a number of individually brazed blocks into a composite assembly. The maximum size of each block is governed by the brazing facilities and techniques. Present

Fig. 5. Contra-flow type 1.

Fig. 6. Contra-flow type 2.

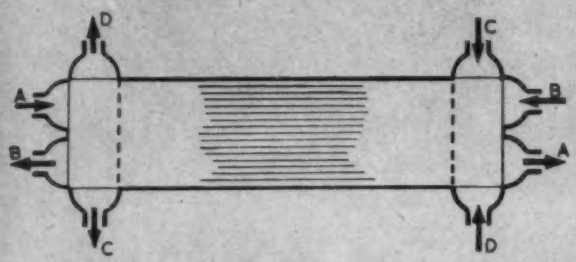


Fig. 7. Contra-flow type 3.

limits for dip-brazed aluminium alloy contra-flow blocks are about 9 ft overall length with 17 in. by 21 in. cross-section, giving a total volume of about 22 cu. ft. A number of such blocks may be assembled together either in parallel or in series, or a combination of both, in order to cater for the overall specified duty. A typical example is shown in Fig. 10. A further possibility is to weld together several separate heat exchangers into a composite assembly, with a common stream on one side and several distinct streams in parallel and series on the other side.

Pressure Limitations

For low to medium steady operating pressures, say, 50-100 psig, the block layout and headering arrangements can be determined primarily on performance and installation considerations, and in general no undue mechanical problems are encountered.

For higher pressures and for reversing pressures, mechanical design aspects must be considered from the outset. The two major factors are the internal pressure loadings on the corrugations, and those on the header tanks, together with any external loadings imposed through the pipework. The strength of the corrugations and associated brazed joints obviously dictates the maximum pressure to which this type of construction may be subjected. Depending upon their type and thickness, the aluminium corrugations are generally satisfactory for static test pressures in the range 600 to 1000 psig. For low-temperature applications the corresponding rated maximum operating pressures would be in the range 250 to 450 psig for steady conditions and 125 to 225 psig for reversing applications. Typical test units have been pressure cycled at room temperature between 0 and 150 psig and have withstood over 1 million reversals without failure. Static test samples have been taken up to 1800 psig.

The pressure loadings on the header tanks, however, present a more immediate problem, for as the test pressure is increased the tank span must be reduced in order to keep the peripheral loading within reasonable limits. The position is further aggravated if additional external loadings are imposed through the adjacent pipework, due to thermal contraction and expansion effects. This means either that the block cross-section must be kept small on high-pressure applications, to suit the tank design requirements, or that small tanks must be fitted, with internal distributors to spread the flow within the block itself, such distributors also being designed to withstand the high pressures involved. Both methods have their uses, depending upon other design considerations.

Large-scale units have been made and operated in this country at steady pressures up to 300 psig with test pressures up to 500 psig at room temperature, while designs of large cross-section blocks have been investigated for test pressures up to 600 psig. Small-scale units could obviously be taken to somewhat higher figures, and developments are in hand to extend the range on the larger units.

Fig. 11 shows a cross-flow heat exchanger with a high-pressure vertical stream and a low-pressure horizontal stream. This unit was made up in eight separate elements owing to the tank stresses on the high-pressure side which was tested to about 400 psig.

Fig. 12 shows a contra-flow assembly fitted with twin end tanks. This type of arrangement is ideally suited to reversing applications owing to the symmetry of the two sets of passages. Under such circumstances, both sets of

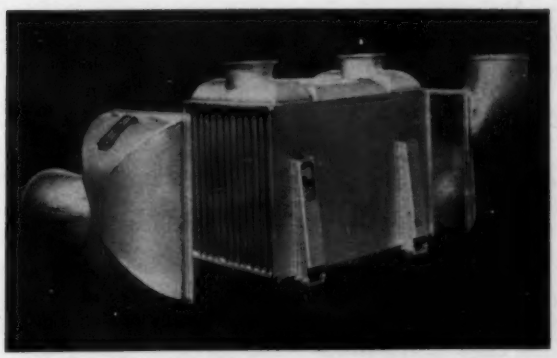


Fig. 8. Cross-contra-flow unit with end tanks attached.

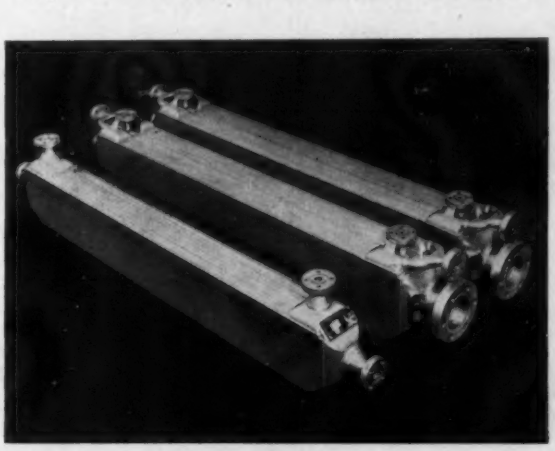


Fig. 9. Heat exchangers for gas purification plant.

passages must be subjected to the same test pressure, which is much higher than for comparable operating pressures under steady conditions. This type of arrangement is also suitable for use with two high-pressure streams, or, with the addition of side tanks, and perhaps a central end tank, for operation with three, four or five streams. Such units have been made and operated extensively in the U.S.A., where the demand developed earlier than in this country.

Fig. 13 shows a small experimental reversing heat exchanger which is one of a set supplied to A.E.R.E. Harwell for their experimental work on hydrogen purification and distillation. This is a particularly interesting illustration of the manufacturing flexibility which can be achieved. Each block contains five separate streams, and the five blocks are connected in series on all five streams. The two main hydrogen streams are reversed between 0 and 10 atma. This unit has been operated between room temperature and liquid nitrogen temperature, while the remaining units have extended the range down to liquid hydrogen temperature. In order to facilitate installation of the units and fitting of instruments, all of the open-ended pipe connections were made with flash-butt welded aluminium-copper pipes as used in the refrigerating industry. These joints have proved to be fully satisfactory at the somewhat ardous operating conditions imposed.

Vacuum Conditions and Tests

The plate-and-corrugation construction is used extensively for operation under partial vacuum conditions. Such units are always subjected to a hydrogen leak detection test at high vacuum on the final assembly. Similar tests are also carried out when required on units designed for operation at positive pressures, thus ensuring an exceptionally high standard of leak tightness.

Performance Estimates and Range of Application

The basic methods of performance calculation are the same as for any other type of heat exchanger, and similarly rely upon experimentally-derived performance characteristics for the type and geometry of surface to be used. Some typical curves for single-phase flow are shown in Fig. 14. Heat-transfer characteristics are plotted as Stanton Number (for air) versus Reynolds Number, the latter being based upon an equivalent hydraulic mean diameter between the corrugations. This does not provide a universal correlation for any particular type of surface but the curves shown represent a close approximation for the corrugations detailed in Table 1. These curves may be used directly for any gas with Prandtl Number similar to that for air. For any other fluids, the curves may be modified to the Colburn factor j' by including the usual Prandtl Number term.

The "multi-entry" and "herringbone" corrugations offer considerably enhanced heat-transfer and pressure-loss characteristics, resulting in a marked reduction in block length required but some increase in total cross-section for a given thermal duty and pressure loss. This is particularly valuable for units involving close temperature approach throughout a wide temperature range, though total effective passage lengths of up to 20 ft may still be required.

Fin Efficiency

Once the heat-transfer coefficient has been determined, the fin efficiency must be evaluated for each stream. The standard relationship for straight uniform fins applies. For ease of application, this is plotted on Fig. 15 in terms of two parameters for direct use with Al-1.2% Mn alloy at room temperature, with suitable correction factors for lower temperatures. These curves may also be used for other metals, or for different fin heights, provided that

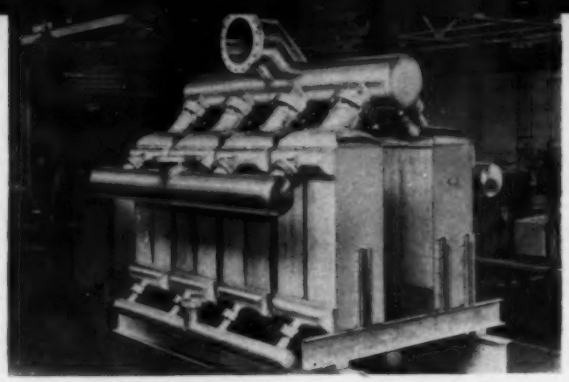


Fig. 10. *Heat exchanger for tonnage oxygen plant.*

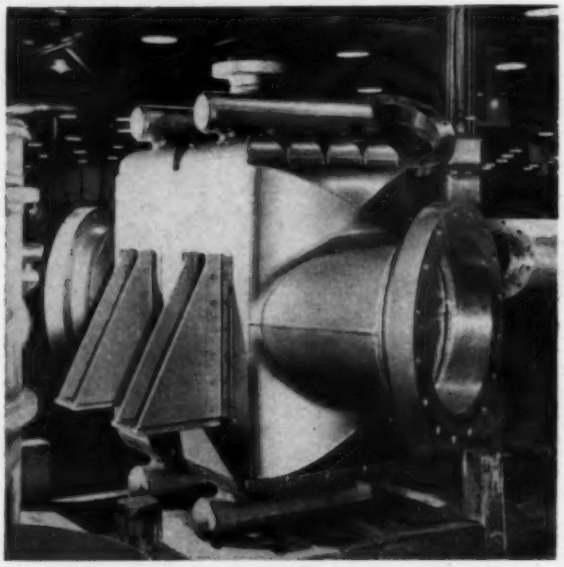


Fig. 11. *High-pressure cross-flow heat exchanger.*

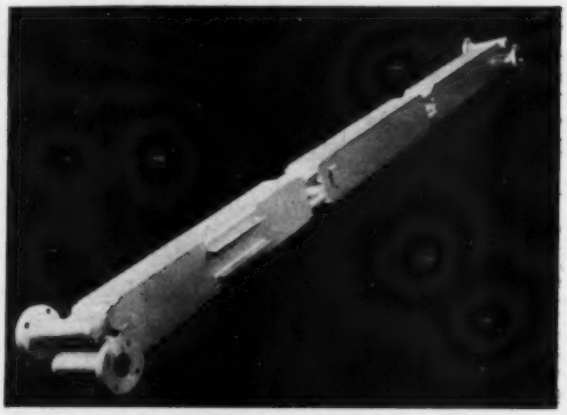


Fig. 12 (*above*). *Two-stream symmetrical assembly.*

Fig. 13. *Experimental unit for hydrogen purification.*

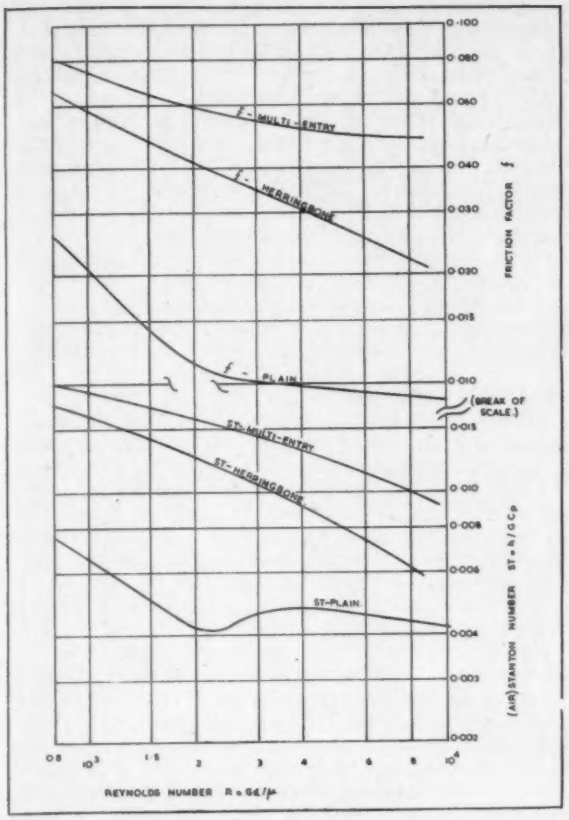


Fig. 14. Performance characteristics for Marston corrugations.

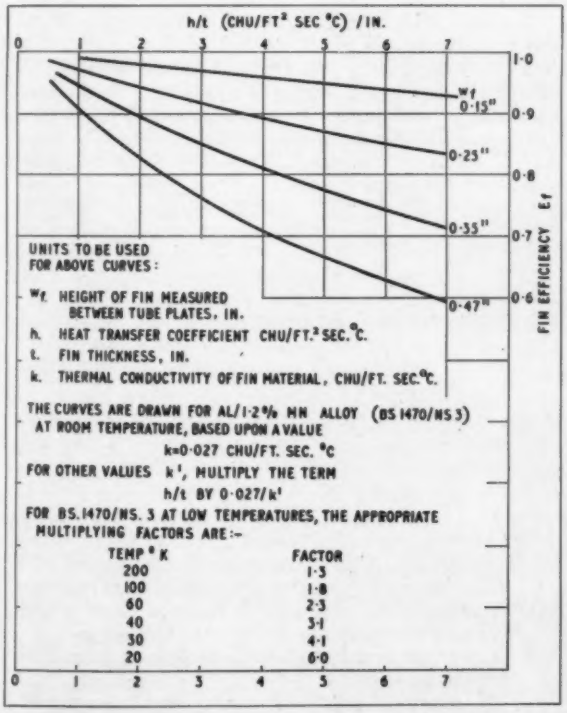

h/t (CHU/FT² SEC °C) /IN.
Wf 0·15"
0·25"
0·35"
0·47"
FIN EFFICIENCY Ef
UNITS TO BE USED FOR ABOVE CURVES:
Wf. HEIGHT OF FIN MEASURED BETWEEN TUBE PLATES, IN.
h. HEAT TRANSFER COEFFICIENT CHU/FT.² SEC. °C.
t. FIN THICKNESS, IN.
k. THERMAL CONDUCTIVITY OF FIN MATERIAL, CHU/FT. SEC. °C.
THE CURVES ARE DRAWN FOR AL/1·2% MN ALLOY (BS 1470/NS 3) AT ROOM TEMPERATURE, BASED UPON A VALUE
k=0·027 CHU/FT. SEC. °C
FOR OTHER VALUES k', MULTIPLY THE TERM
h/t BY 0·027/k'
FOR BS.1470/NS. 3 AT LOW TEMPERATURES, THE APPROPRIATE MULTIPLYING FACTORS ARE:-
TEMP ° K FACTOR
200 1·3
100 1·8
60 2·3
40 3·1
30 4·1
20 6·0

Fig. 15. Fin efficiency for straight uniform fins.

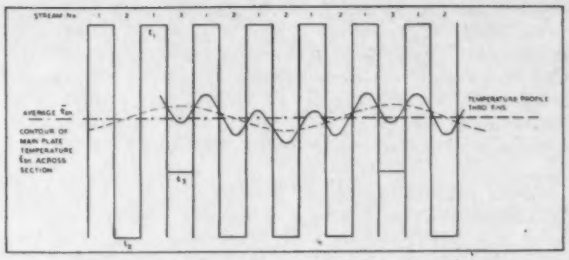


Fig. 16. Temperature distribution diagram.

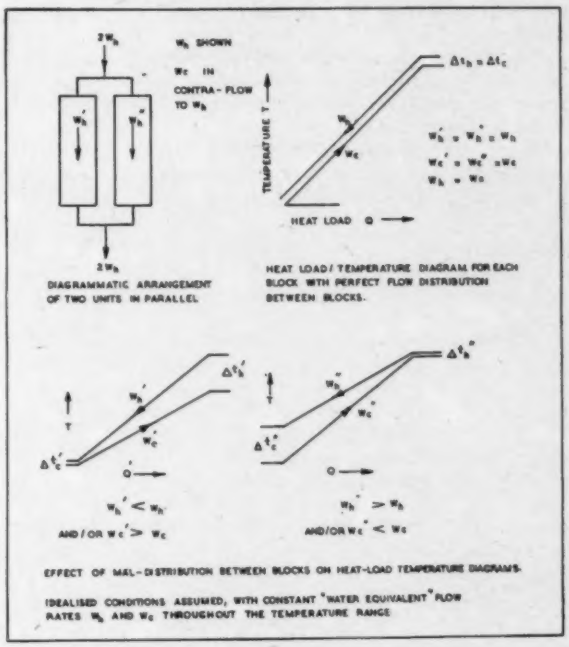

2Wh
Wh SHOWN
Wc IN CONTRA-FLOW TO Wh
DIAGRAMMATIC ARRANGEMENT OF TWO UNITS IN PARALLEL
TEMPERATURE T
HEAT LOAD Q
HEAT LOAD / TEMPERATURE DIAGRAM FOR EACH BLOCK WITH PERFECT FLOW DISTRIBUTION BETWEEN BLOCKS.
Wh' < Wh
AND/OR Wc' > Wc
Wh" > Wh
AND/OR Wc" < Wc
EFFECT OF MAL-DISTRIBUTION BETWEEN BLOCKS ON HEAT-LOAD TEMPERATURE DIAGRAMS.
IDEALISED CONDITIONS ASSUMED, WITH CONSTANT "WATER EQUIVALENT" FLOW RATES Wh AND Wc THROUGHOUT THE TEMPERATURE RANGE

Fig. 17. Heat load temperature diagrams.

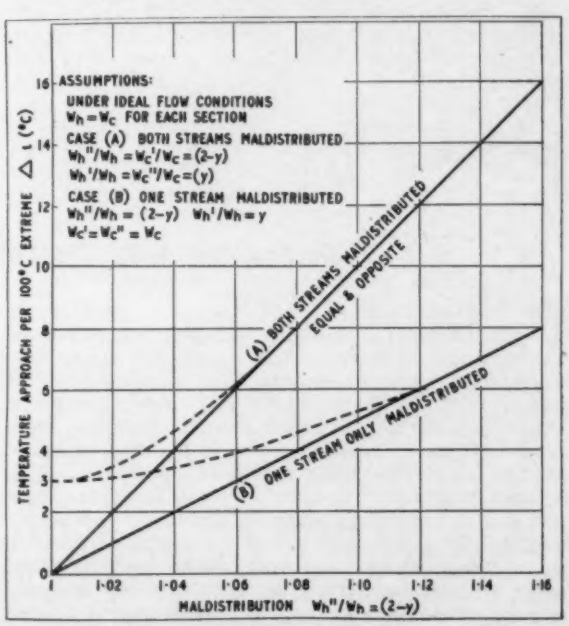

ASSUMPTIONS:
UNDER IDEAL FLOW CONDITIONS
Wh = Wc FOR EACH SECTION
CASE (A) BOTH STREAMS MALDISTRIBUTED
Wh"/Wh = Wc'/Wc = (2-y)
Wh'/Wh = Wc"/Wc = (y)
CASE (B) ONE STREAM MALDISTRIBUTED
Wh"/Wh = (2-y) Wh'/Wh = y
Wc' = Wc" = Wc
TEMPERATURE APPROACH PER 100°C EXTREME Δt (°C)
(A) BOTH STREAMS MALDISTRIBUTED EQUAL & OPPOSITE
(B) ONE STREAM ONLY MALDISTRIBUTED
MALDISTRIBUTION Wh"/Wh = (2-y)

Fig. 18. Limiting temperature approach with maldistributed flow.

suitable adjustments are made to the two parameters.

In cases where a fouling factor is required on any stream, the effective heat-transfer coefficient including fouling should be evaluated before estimating the appropriate fin efficiency. The fin efficiency term is applied to the secondary, or fin, surface only; hence the total effective surface per unit length in each stream is obtained.

Overall Heat-transfer Rating

In evaluating the overall heat-transfer rating, it is preferable to depart slightly from standard practice, and for contra-flow blocks to consider the term $(hA)/L$ for each stream where h is the heat-transfer coefficient and A/L is the effective surface area per unit length for the stream considered.

The thermal resistance across the primary plates can generally be ignored for the plate thicknesses and heat loads employed. Hence the overall heat-transfer rating per unit length is obtained as:

$$L/(hA)_{ov} = L/(hA)_h + L/(hA)_c \quad \ldots (1)$$

From the total heat load, Q, and the mean effective temperature difference, $\triangle t$, the required duty is:

$$Q/\triangle t = (hA)_{ov} \quad \ldots (2)$$

from which follows the total net length, L.

This may be increased to include a design margin. Appropriate allowances must also be made to cover end distributors within each block.

If more than two separate streams are included in a single heat exchanger, there is a further consideration which may be significant and where the treatment may differ from that used for tubular constructions. Since the whole unit is brazed into a single "honeycomb" structure, there must be continuity of metal temperature throughout. Hence, if several separate cooling streams with similar heat-transfer characteristics but at somewhat different initial temperatures are distributed across the section, their temperatures will tend to converge to the same outlet value.

Fig. 16 shows a typical temperature distribution diagram across a section of a three-stream heat exchanger. Under these conditions, the heat transfer per passage to stream 3 will be less than to stream 2, and so the local temperature rise per unit length will be less in stream 3 than in stream 2. Thus the difference in temperature levels and resultant relative heat-transfer rates to streams 2 and 3 will be progressively reduced along the length of the unit until completely eliminated.

Flow Distribution

When very close temperature approaches are required, uniformity of flow distribution becomes an important consideration, both within each heat exchanger element and, for composite assemblies, between individual elements. Fig. 17 illustrates the effect on heat load-temperature diagrams of flow mal-distribution between two elements in parallel, while the quantitative effect on overall performance is given in Fig. 18 for the limiting design condition of zero temperature difference, i.e., infinite heat exchanger length, and for a design approach of 3°C per 100°C extreme between hot and cold inlet temperatures. Considerable theoretical and experimental work has been carried out on this problem regarding both internal distributors and external manifolding. While pressure losses are kept as low as possible in the external manifolding, those in the internal distributors are inevitably relatively high, since these distributors must also be designed to carry the internal pressure loading across the primary plates.

Range of Application

From the performance point of view, the secondary-surface plate-and-corrugation type of construction offers an attractive solution to almost any type of heat exchanger problem in the low-temperature field, covering any combination of gaseous, liquid, condensing or evaporating streams. The type and arrangement of corrugations can be selected to suit each particular application. The maximum size and pressure limitations have been set out above. Compatability of the working fluids and materials of construction must always be considered.

Problems of fouling and possible blockage should be investigated at an early stage, first to establish the suitability of the application for this type of construction, and, secondly, to ensure that the most appropriate type of corrugation and layout is adopted. For instance, in reversing heat exchangers in which removal of solid impurities is required, the "multi-entry" corrugation is effective in retaining solid particles at their point of deposition. In other cases, plain passages may be preferred to permit free transit of contaminating solids through the heat exchanger. In the case of "permanent" fouling, chemical cleaning methods can sometimes be used.

Figs. 8 to 13 serve to illustrate the wide range of design flexibility which can be provided. Variations in size are almost infinite. The largest single composite assembly so far manufactured and operated contains no less than 160 cu. ft of effective heat-transfer volume. The smallest aircraft unit made is a mere 3-in. cube.

ACKNOWLEDGMENTS

Acknowledgments are due to the directors of Marston Excelsior Ltd. for permission to present this paper, and to all who have assisted in its preparation; to British Oxygen Engineering Ltd. for permission to include Figs. 9, 10 and 11 illustrating some of the units supplied for their plants; and to A.E.R.E., Harwell, for permission to include Fig. 13.

Operating and Testing Connersville Meters

CONNERSVILLE meters are based on the design of the "Roots blower", except that the impellers are of a more exact epicycloidal profile and are machined to a very close tolerance. At the 25th Autumn Research meeting of the Institution of Gas Engineers, G. Dougill presented a report (Publication No. 557) on their operation and testing, based on over 20 years of experience. Tests of accuracy were based on both calculations of the gas flow assessed by the speed of rotation and the known swept volume of the impellers, and also on comparison against standardised wet meters.

The Connersville meters proved to be satisfactory, usable up to 30% overload and even in one test up to 200% of the rated load. Apart from bearing failure, the most frequent trouble is lack of exact synchronisation of the two impellers. The mechanical clearances are small and the impellers must never come into contact at any part of the revolution. To achieve this, the back-lash of one pair of gears is taken up in the clockwise sense, and the back-lash of the other pair is taken up in the counter-clockwise direction, whilst the clearance at each side of the impellers is equally divided.

Mr. Dougill stated that it is now standard practice for all North Thames new meters, and also for any meters removed from their foundations for overhaul, to undergo the following tests and examinations before being installed:

(1) All mechanical clearances are measured and recorded.
(2) A volumetric accuracy test is taken at 10, 20, 30, 40, 70 and 100% of full load.
(3) A slip test is taken and recorded in rpm at three differential pressures.
(4) The counter is checked to confirm that it is operating properly and is of the correct ratio.

662.995

A SIMPLIFIED REGENERATOR THEORY

The derivation of a simple expression for regenerator efficiency is discussed by the author

by W. F. SCHALKWIJK, Ph.D.

THE possibility of a simple formula for η, the regenerator efficiency, arises from the fact that most regenerators encountered in practice have very simple temperature fields in their central parts. In the symmetrical case, which we treat here, this temperature field is to a high degree of approximation linear, viz., equal to the zero'th Eigenfunction introduced by HAUSEN.[1] This Eigenfunction is the simplest solution of the differential equations, and can be written for the matrix temperatures as follows:

$$T_w = A + B(x - t)$$
$$T'_w = A + B(x - \Pi + t) \qquad 0 \leqq x \leqq \Lambda,\ 0 \leqq t \leqq \Pi$$

T_w represents the matrix temperature during the period in which the gas flows in the positive direction (from the low-temperature side to the high-temperature side), T'_w is the matrix temperature during the negative period, A and B are constants, and Λ and Π are the dimensionless length and period of the regenerator respectively.

The higher Eigenfunctions decrease rapidly towards the centre of the regenerator, as shown in Fig. 1, where the matrix temperature is represented at the beginning and the end of a period by the full lines, whereas the zero'th Eigenfunction for the same times is represented by a dotted line. In the central part of the regenerator the lines coincide. The gas temperature difference across the regenerator is denoted by $\triangle T_g$. If the temperature field in the central part of the regenerator is linear we can draw from this fact two important conclusions.

The heat loss of a regenerator during a complete cycle is equivalent to an amount of enthalpy (or heat) which enters the regenerator at the high-temperature side, is passed on unchanged through every cross-section of the regenerator (according to the first law of thermodynamics), and leaves the regenerator at the low-temperature side. This "enthalpy transport" is due to the *gas* having a different mean temperature in both periods.

We shall indicate this enthalpy transport by H. As it must have the same value in every cross-section, we can calculate its value at any point we like. We start with a point where only the zero'th Eigenfunction is present. By calculating the gas temperatures from the matrix temperatures and averaging, we find easily:

$$H = 2BC_g.$$

C_g represents the total thermal capacity of the gas flowing through the regenerator during the positive or the negative period.

From the definition of efficiency:

$$H = (1 - \eta)\,\triangle T_g C_g.$$

Dr. Schalkwijk *is with Philips Research Laboratories, N.V. Philips' Gloeilampenfabrieken, Eindhoven-Netherlands.*

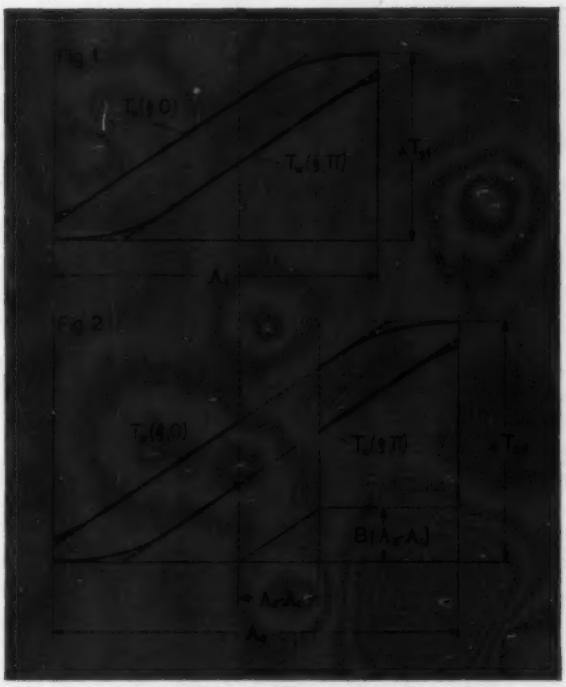


Fig. 1 (top). Temperature field in regenerator.
Fig. 2. Temperature field in longer regenerator of the same class.

From both formulæ one finds, as the quantity C_g cancels:

$$(1 - \eta)\,\triangle T_g = 2B \qquad \ldots . (1)$$

This is our first result, derived from the fact that the temperature field in the central part is linear.

We now compare *two* regenerators with different values of Λ and $\triangle T_g$ *but the same value of* Π. We can prove rigorously that in this case, and *only* in this case, we can make the zero'th Eigenfunctions in both regenerators identical by choosing suitable values for $\triangle T_g$ in these regenerators. This is shown in Fig. 2. If we compare this with Fig. 1, we see that the right-hand part of the regenerator is slid upwards and to the right along the dotted lines until it occupies the position indicated in Fig. 2. In this way we have constructed the matrix temperature field which must be present in the second regenerator.

From Fig. 2 we can derive the relationship between both *gas* temperature differences. We draw a line parallel

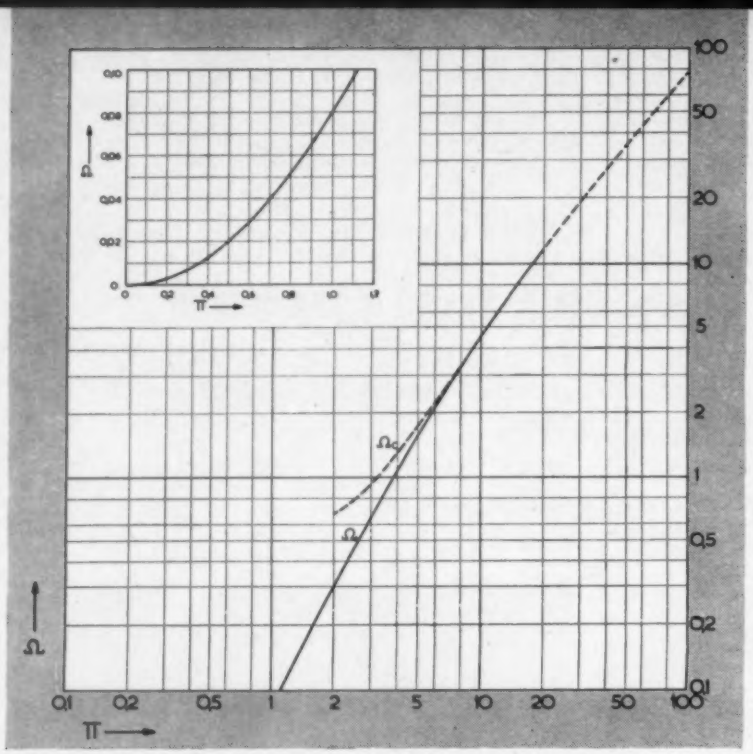


Fig. 3. Graph indicating the values of Ω *(and* Ω_c*) as a function of* Π.

to the dotted lines (see Fig. 2) and see at once that the following relation must hold:

$$\Delta T_{g2} - \Delta T_{g1} = B(\Lambda_2 - \Lambda_1) \qquad \ldots . (2)$$

This is the second result we need.

Employing Equation (1) allows us to eliminate ΔT_{g1} and ΔT_{g2} from Equation (2). The quantity B cancels, as it has the same value in both regenerators. After some rearranging we obtain:

$$\Lambda_1 - \frac{2}{1-\eta_1} = \Lambda_2 - \frac{2}{1-\eta_2}.$$

As this relation must hold for every value of Λ_2, we see that this expression is invariant for a whole class of regenerators having the same value of Π. For regenerators with different values of Π, the expression also has different values.

Hence we may write:

$$\Lambda + 2 - \frac{2}{1-\eta} = \Omega\,(\Pi)$$

when Ω is a correction to Λ and depends only on Π.

We added the constant 2 to the expression, which is, of course, permissible, and which ensures that Ω always has positive values. This can be restated in the form:

$$\eta = \frac{\Lambda - \Omega}{\Lambda - \Omega + 2}, \quad \Omega = \Omega\,(\Pi).$$

We now find, if we calculate Ω from known values of η, that it is indeed independent of Λ (or Γ) if we keep Π constant. In the following small table we give slightly rounded-off figures for Ω, Π being equal to 4.

Γ	1	2	3	4	5
Ω	0.99	1.07	1.07	1.07	1.07

Already for $\Gamma = 2$ we see that Ω becomes constant. This has been verified for values of Π between 1 and 20. The limiting values of Ω have been determined as correctly as possible from known values of η for the given values of Π (cf. ref. 2). These values are shown in the following table.

Values of Π for different values of Ω

Π	Ω	Π	Ω	Π	Ω	Π	Ω
1	0.081	6	2.07	11	5.12	16	8.58
2	0.31	7	2.63	12	5.79	17	9.30
3	0.65	8	3.22	13	6.48	18	10.03
4	1.07	9	3.84	14	7.17	19	10.76
5	1.55	10	4.47	15	7.87	20	11.50

To extrapolate the values of Ω to higher values of Π we used the formula:

$$\Omega_c = \Pi + 2 - 2.35\sqrt{\Pi}.$$

This formula is to be considered as an asymptotic expression which holds true up to the highest values of Π, but which is not valid for low values of Π. This is indicated in Fig. 3, where the calculated values of Ω are shown by a full line and the asymptotic formula by a broken line. They are seen to coincide in the vicinity of $\Pi = 20$.

The accuracy of our representation was tested by calculating values of $(1 - \eta)$, as this quantity is generally more important than η itself. For $\Gamma = 1$, and values of Π between 1 and 20, relative errors of the order of magnitude of $\sim 1\%$ were found. For $\Gamma = 2$ the relative error quickly became insignificantly small. As in practice $\Gamma > 1$ will always hold, it will be found that our formula is sufficiently accurate to represent all practical cases.

An analogous theory can also be developed for the general case, in which the parameters have different values in both directions. Here, also, a reduction of the number of parameters is achieved by the introduction of an invariant expression.

REFERENCES

[1] Hausen, Helmuth "Wärmeübertragung in Gegenstrom, Gleichstrom und Kreuzstrom", Springer-Berlin, Göttingen, Heidelberg, 1950, 406-12.

[2] Lambertson, T. J. "Performance Factors of a Periodic Flow Heat Exchanger." Paper presented at the Semi-Annual Meeting, San Francisco, California, June 9-13, 1957, of the A.S.M.E. Paper No. 57-Sa-13.

662.995 : 66.012.7

by H. LONDON

GENERAL CONSIDERATIONS ON EFFICIENCY AND COSTS OF REGENERATORS

This paper sets out to answer the question "Is there a theoretical limit to the efficiency of a heat exchanger?", and to show also how an increase in efficiency is related to an increase in capital cost.

THE purpose of a regenerator is to exchange the enthalpy of two gas streams, the temperatures of which are increased or decreased by $\triangle T$ such that the mean temperature difference, $2\overline{\delta T}$, between the two streams is as small as possible. This should be achieved with the pressure drop, $\triangle p$, along each gas stream kept as small as possible. This pressure drop will, in general, be different for the two streams if the mean pressure of the streams differs. But since we intend to apply only dimensional arguments, we shall assume equal pressures for both streams; in the case where the two streams differ substantially in pressure, the pressure drop in the low-pressure stream would be the determining factor.

A regenerator can be related to a balanced two-stream countercurrent recuperator of equal heat-transfer characteristics. The average temperature difference between the two streams of the regenerator, $2\overline{\delta T}$, is then related to that of the corresponding recuperator, $2\delta T$, by an efficiency factor

$$\eta = \delta T/\overline{\delta T} \qquad \ldots . (1)$$

which depends mainly on the amplitude of the temperature oscillation $\gamma\triangle T$ of the central part of the regenerator. In a regenerator in which $\overline{\delta T}/\triangle T \ll 1$, the efficiency factor η is approximately given by

$$\eta = 1 - 0.7\gamma \qquad \ldots . (2)$$

This expression represents fairly well results originally due to HAUSEN and given by JAKOB,[1] and also some data referring to still smaller $\overline{\delta T}/\triangle T$ which have been made available to me by RUHEMANN.

We describe the heat-transfer process in a volume element of a regenerator or equivalent recuperator in terms of the transfer length l, which corresponds to the height of a theoretical plate in distillation. It is defined as the distance in the direction of the gas stream over which the temperature changes by an amount equal to the difference in temperature between gas and packing.

The transfer length is given by

$$l = m/St \qquad \ldots . (3)$$

where m is the hydraulic radius, i.e., the ratio of total volume to total surface of the packing and

$$St = h/GC_p \qquad \ldots . (4)$$

is the Stanton number, h being the heat-transfer coefficient, G the mass flow per total cross-section and C_p the specific heat per unit mass at constant pressure. St is a function of:

$$\mathrm{Re} = mG/\mu$$

where μ is the viscosity of the gas.

For a packed bed and comparatively large Re, DENTON[2] has found

$$St = 0.48\ (\mathrm{Re})^{-0.3} \qquad \ldots . (4a)$$

For laminar flow $St \propto (\mathrm{Re})^{-1}$ .... (4b)

Thus, in general, $l \propto m^n$ .... (5)

where $1.3 < n < 2$ .... (5a)

These expressions for the transfer length l take account only of the gas to surface heat transfer. The heat resistance of the solid, which is negligible, and heat conduction in the direction of the gas flow, which can be important, can be formally expressed as an addition to (1). This is taken into account in the paper in (14).

If we denote by L the total length of the exchanger and by

$$N = L/l \qquad \ldots . (6)$$

the number of transfer stages, also called the reduced length, then it follows from the definition of l that the thermal effectiveness

$$\theta \equiv 2\overline{\delta T}/\triangle T = 2/\eta N \qquad \ldots . (7)$$

We next consider the pressure drop, δp, along the transfer length, l. We express it in terms of the Fanning friction factor f:

$$\delta p = l\,.\,f\,.\,G^2/2\rho m \qquad \ldots . (8)$$

where ρ is the density.

With (3), this becomes

$$\delta p = (f/St)\,G^2/2\rho = (f/St)\rho v^2/2 \qquad \ldots . (9)$$

The second form contains v, the linear velocity referred to the total cross-section. For gases the ratio f/St depends little on the Reynolds number. It therefore depends mainly on the geometry, not on the absolute dimension of the system. For open structures, e.g., thin-walled tubes, the order of magnitude f/St is 1, and it can be said that in any reasonable heat-transfer system involving gases, the pressure drop over the transfer length is of the order of magnitude of the velocity head $\frac{1}{2}\rho\overline{v_l^2}$, where v_l denotes the maximum local velocity with the mean velocity taken along the lines of flow. The logarithmic pressure drop is

$$\frac{\delta p}{p} = (f/St)\,v^2/(2RT/M) = \frac{3}{2}(f/St)\,(\overline{v^2}/\overline{v^2}_{\mathrm{mol}}) \approx \frac{3}{2}\overline{v_l^2}/\overline{v^2}_{\mathrm{mol}} \qquad \ldots . (10)$$

Here we have introduced the mean square of the molecular velocity

$$\overline{v^2}_{\mathrm{mol}} = 3RT/M \qquad \ldots . (11)$$

where R is the gas constant and M the molecular weight. Thus we have the interesting result that the relative pressure drop over the transfer length is of the order of magnitude of the square of the ratio of the local velocity to the r.m.s. gas kinetic velocity.

The total relative pressure drop over the two regenerators (if the two gas streams have the same pressure) is

$$\pi \equiv \frac{2\triangle p}{p} = 3(f/St)\,(\overline{v^2}/\overline{v^2}_{\mathrm{mol}})N = 6(f/St)\,(\overline{v^2}/\overline{v^2}_{\mathrm{mol}})/(\eta\theta) \qquad \ldots . (12)$$

or $$\pi = (f/St)\,(M/R)\,(\overline{1/T\rho^2})\,G^2N \qquad \ldots . (12a)$$

where the mean is taken along the length of the regenerator, say, during the middle of a period.

We note that for a given geometry, i.e., a given value of (f/St) the pressure drop depends only on gas velocity and on the thermal effectiveness, θ, or on the number of transfer stages, N. It does not depend on m, the hydraulic radius of the channels, and we can make the pressure drop as small as we like if we give the regenerator a sufficiently large area.

A feature peculiar to regenerators which sets a limit to their usefulness at very low temperatures and very high

Mr. London *is with the Atomic Energy Research Establishment, Harwell.*

pressures is the ratio of gas hold-up to the amount of gas passed during a period. This is

$$r = \frac{w\rho C_p}{(1-w)\rho_s C_s} \quad \ldots.(13)$$

where w is the voidage of the packing and ρ_s, C_s are the density and specific heat of the packing material.

This expression is approximate in as far as it assumes infinite conductivity inside the packing, but with $\gamma > \theta$ the correction is small, and it works in the way that an improvement in θ (i.e., decrease of θ) will decrease r. How small r has to be depends on the manner one can deal with the gas hold-up when switching the regenerators in any particular application. What matters here is that r depends only on the physical properties of gas and packing, and on the geometry, but neither on the efficiences θ and π nor the hydraulic radius m. It is true that both θ ((2) and (7)) and r depend on the temperature oscillation γ, but this means that one has to fix γ so that r in equ. (13) is as small as necessary and then improve θ by increasing the length of the regenerator and the number of transfer stages, N. The increase in length does not affect r, as one might have thought, because it leads to a longer period for the same amplitude of temperature oscillation. Thus, although the hold-up goes up, the amount of gas passed increases by the same factor.

We have now established that we can reduce both the thermal losses and the pressure losses as much as we are prepared to pay for, by simply increasing the length and area of the regenerator without any other considerations entering. But before we can establish a relation between size and efficiency we have still to settle the approximate value of the hydraulic radius m to which the length of the regenerator is related through the transfer length l, which increases with m according to (5). As m has no effect on either π or r, it looks at first that one should make m as small as possible, and here the advantage of the regenerator over the recuperator is evident. It is easier technically to make a finely divided bed than it is to fabricate a multiplicity of separate channels which must lead to separate headers. It is also not important to obtain an even flow distribution. The flow of the opposing stream takes place through the same channels and with the same Reynolds number. Therefore, the local deviations from the average will be the same and the two streams will be, so far as their flow is concerned, locally balanced.

There is, however, a lower limit to the reduction in l imposed by the longitudinal heat conduction. This leads to an additional term to l in (3), leading to the amended value

$$l^* = l + K/GC_p = (m/St)(1 + K/GC_p l) \quad \ldots.(14)$$

where K is the combined gas and solid thermal conductivity of the bed in the longitudinal direction. We cannot, therefore, reduce the transfer length indefinitely:

$$l^* > K/GC_p \quad \ldots.(15)$$

One can show that the minimum volume of the regenerator is obtained if

$$l = K/GC_p \quad \ldots.(16)$$

so that

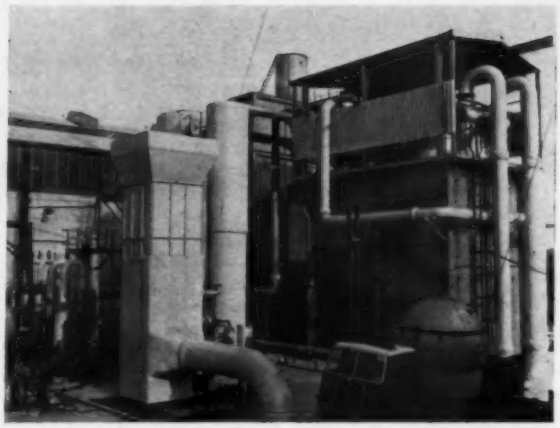
$$l^* = 2K/GC_p = 2l \quad \ldots.(17)$$

Then (12a) has to be amended into

$$\pi^* = 2\pi = 2(f/St)(M/R)(\overline{1/T^2})\,G^2/N \quad \ldots.(18)$$

From this point of view, it is preferable to use a packing of low thermal conductivity, i.e., ceramics or pebbles. Even these have a considerably higher thermal conductivity than the gas. Thus their effect on the heat transfer inside the packing should not be too unfavourable.

Equation (15) tells us that if we reduce G in order to reduce the pressure drop we have to increase the transfer length and the size of the packing elements. As the cross-sectional area for a given throughput is also inversely proportional to G, we find that the volume of a regenerator of optimum particle size is proportional to G^2



Britain's first tonnage oxygen plant at Margam, near Swansea. The plant is the first of five which are being constructed to service individual iron and steel works in many parts of the country. (Photo: B.O.C. Ltd.)

The various cost items per unit mass flow for a pair of regenerators can now be considered. If Θ is the refrigeration or heating cost per power unit, we have the thermal cost per unit mass flow (7)

$$\pounds_\theta = \Theta\theta C_p \triangle T = 2\Theta C_p \triangle T/\eta N \quad \ldots.(19)$$

If Π is the running and capital cost per unit compressed power, we have the compressed power cost per unit mass flow (18):

$$\pounds_\Pi = \Pi\pi^* RT_r/M = 2\Pi(f/St)\,T_r(\overline{1/T\rho^2})\,G^2 N \quad \ldots.(20)$$

where T_r is room temperature. Finally, let Γ be the capital charge per unit volume of the regenerator. Then this charge per unit mass flow is (15):

$$\pounds_\Gamma = \Gamma V = \Gamma l^* N/G = 2\Gamma(K/C_p)N/G^2 \quad \ldots.(21)$$

The total cost per unit mass flow is therefore

$$\pounds = \pounds_\theta + \pounds_\Pi + \pounds_\Gamma \quad \ldots.(22)$$

This cost has to be minimised with respect to the only two variables, N and G, by making

$$\partial\pounds/\partial N = 0 \text{ and } \partial\pounds/\partial G = 0$$

The result is: $G^4 = \Gamma(K/C_p) \,/\, \Theta(C_p \triangle T/\eta) \quad \ldots.(23)$

$$N^4 = \tfrac{1}{4}(\Theta C_p \triangle T/\eta)^2/[\Pi(f/St)\,T_r(\overline{1/T\rho^2}).\,\Gamma(K/C_p)] \quad \ldots.(24)$$

$$\pounds_{min} = 2\sqrt{2}\,(\Theta C_p \triangle T/\eta)^{\frac{1}{2}}[\Pi(f/St)\,T_r(\overline{1/T\rho^2})]^{\frac{1}{2}}[\Gamma(K/C_p)]^{\frac{1}{2}} \quad \ldots.(25)$$

$$\pounds_\theta : \pounds_\Pi : \pounds_\Gamma = \tfrac{1}{2} : \tfrac{1}{4} : \tfrac{1}{4} \quad \ldots.(26)$$

Thus the volume dependent capital cost should be $\frac{1}{4}$ of the combined power cost and the thermal part of the power should cost twice as much as the mechanical power. The expression shows also how economies on any of the three items affects the total cost. Once the best mass flow rate has been calculated from (23), the appropriate particle size m of the packing can be determined by means of (15), (3) and (4a), or the equivalent of (4a) appropriate to any packing of different shape.

REFERENCES

[1] Jakob, M. "Heat Transfer", Vol. II, p. 295, Table 35-5, John Wiley & Sons, 1957.

[2] Denton, W. H. Inst. Mech. Eng., Heat Transfer Discussions, 1951, p. 370.

66.045.2 : 66.048.2

HEAT TRANSFER ACCOMPANYING THE CONDENSATION OF MIXED VAPOURS

A review of existing methods of calculating condensing film coefficients for mixed vapours

by G. G. HASELDEN,* Ph.D., and W. A. PLATT,* M.Sc.

WHILE there are many instances of mixed vapour condensation in the chemical industry, most of these applications appear to occur in conditions where large temperature differences are available and high accuracy of design is unnecessary. Recent developments have, however, posed the problem of achieving high rates of heat transfer with small temperature differences, and in these circumstances the designer finds little guidance as to either the best form of condenser to use or a reliable method of design. This problem becomes most acute in the refrigeration and low-temperature industry, not only because the achievement of low power consumption requires low-temperature heads, but also because penalties arising from employment of excess heat-transfer surface are high.

The purpose of this paper is to assess the available methods for calculating heat-transfer coefficients for mixed vapour condensation, to discuss the required design features of an efficient mixed vapour condenser and to report some experimental results for one recommended type.

* Chemical Engineering Department, Imperial College of Science and Technology, London, S.W.7.

Effect of Vapour and Liquid Flow Pattern on the Condensation Process

Although the point is not generally stressed, the type of condensation process occurring with a mixture is strongly influenced by the vapour and condensate flow patterns. With some flow patterns large temperature heads are encountered which contribute nothing to the process of heat transfer. Three possible flow patterns for the total condensation of a binary vapour mixture are illustrated in Fig. 1. In type (a) vapour is assumed to approach a cooled vertical surface along a normal. If total condensation occurs the condensate will have the same composition as the bulk vapour. However, at the vapour-liquid interface the vapour composition can be expected (except at very high rates of condensation) to have a composition closely approaching that in equilibrium with the conden-

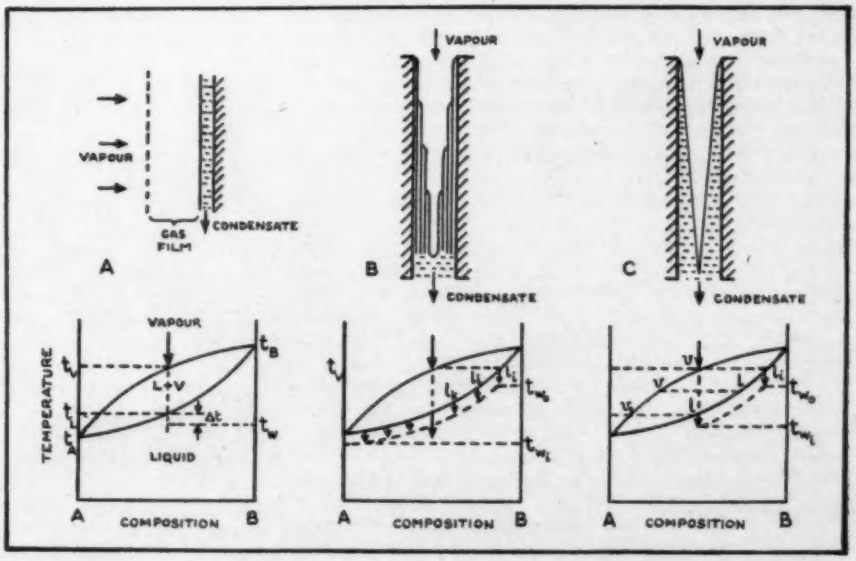


Fig. 1. Three modes of condensation of a binary mixture. (a) One-stage total condensation. (b) Co-current condensation with no liquid mixing. (c) Co-current condensation with liquid mixing.

sate. Since in general this equilibrium composition will differ from the bulk vapour composition there must be a layer or film of vapour adjacent to the interface in which a gradient of composition exists, the thickness of the film being a function of the rate of gaseous diffusion of the more volatile component. A temperature gradient ($t_v - t_L$ in Fig. 1a) will also exist across the vapour film; thus the total temperature difference necessary to effect heat transfer will be the sum of ($t_v - t_L$) and $\triangle t$, the temperature difference across the condensate film given by Nusselt's equation.

In type (b) the vapour flows down through a cooled tube and is progressively condensed, the vapour core being completely mixed but the condensate unmixed at every level. In the accompanying temperature-composition diagram the temperature distribution during condensation is shown. Condensation will continue until the aggregate composition of all the deposited condensate layers is equal to the vapour feed composition. In general, the final quantity of liquid depositing will be nearly pure low boiling component, and heat must be extracted from the condenser over a wide temperature range. If account is taken of resistance in the vapour phase an additional temperature head will be introduced, leading to a lower heat-transfer coefficient.

Condensation according to type (c) represents the ideal condensation process, since heat is liberated from the system at the highest possible temperature, and therefore the temperature difference between it and the coolant is a maximum. This type of condensation requires complete mixing in both phases and also good contact between the phases.

If condensation occurs in the shell of a tubular condenser similar variations in the mechanism of condensation are possible. With a vertical unbaffled unit operating with a large temperature difference the mechanism will approximate to type (a). The same unit with a baffled shell and the vapour flowing generally downwards will approach type (b). The equilibrium, type (c), condensation could not, in general, be approached unless some method of turbulence promotion was incorporated on the tube surface.

Review of Existing Methods of Calculation

The relation of the flow mechanism to the achievable heat-transfer coefficient has been disregarded by most authors. KERN[3] gives the fullest text-book treatment of the problem, but his recommended method of approach disregards the presence of the vapour film altogether and calculates a heat-transfer coefficient based solely on the condensate film resistance by substituting liquid mixture properties in the Nusselt equation. The method can therefore be expected to give misleadingly high values of the coefficient, the error being greatest at low values of temperature difference.

PRESSBURG and TODD[5] employed the opposite approach and considered the condensation of a vapour mixture to take place according to the type (a) mechanism. They showed that this method works experimentally for condensation on the outside of a single horizontal tube. However, the high value of the heat-transfer coefficient obtained in their method by using the bubble point of the vapour rather than its actual temperature is illusory; the effective heat-transfer coefficient is very low due to the thermodynamically irreversible processes occurring in the vapour film. This method is advocated by MCADAMS.[4]

The classical approach was given by COLBURN and DREW,[2] but while their paper is quoted widely it in fact only provides a method for predicting the composition of the first quantity of condensate depositing in the entry region of a condenser. The extension of their approach to deal with a condenser of finite length was due to VAN ES and HEERTJES,[6] who considered the case of condensation of a vapour mixture in a short vertical tube. The equations they derived are, however, restricted to examples in which the amounts of condensation are sufficiently small for the change of composition of the vapour and the condensate over the length of the tube to be negligible. Within this serious limitation they showed that at high vapour velocities the condensation approached the equilibrium condition (type (c), Fig. 1), whereas at low velocities it approached the one-stage process (type (a), Fig. 1). They did not extend their method to the case of complete or nearly complete condensation in which the problem of liquid mixing must be allowed for.

Practical suggestions were given by COLBURN[1] to deal with the case of total condensation of a vapour mixture within a tube, the coolant inlet temperature being below the boiling point of the most volatile component present. He advocated taking the temperature of the vapour/liquid interface as equal to the bubble point of the vapour at the top (inlet) of the tube and equal to the boiling point of the most volatile component (or azeotrope, if formed) at the bottom. This method can be expected to provide a conservative solution.

The application of some of these methods to the condensation of a methanol (71%)/n-propanol (29%) mixture is illustrated in Fig. 2. In this diagram the effective heat-transfer coefficient $Q/(t_v - t_w)$, where Q is the calculated heat flux per unit area and $t_v - t_w$ the difference between the bulk vapour temperature and the mean wall temperature, is plotted against $t_v - t_w$.

For large temperature differences the predictions of the various authors converge, but at values below, say, 50°F, they are incompatible. The curve derived from KERN'S method must be regarded as fictitious in this region, whilst the curves of MCADAMS and COLBURN are too conservative for a well-designed condenser. No satisfactory alternative design method exists at the present time.

Optimum Design for Mixed Vapour Condensation

Although a quantitative design method for an efficient mixed vapour condenser does not exist, it is possible to specify the desirable features to be incorporated in such a condenser. To minimise the thermal resistance of the condensate layer, vapour drag should be used to aid the drainage of condensate. Hence it is best to have the vapour flowing vertically downwards through the condenser. For the same reason, it is beneficial to employ a small vertical

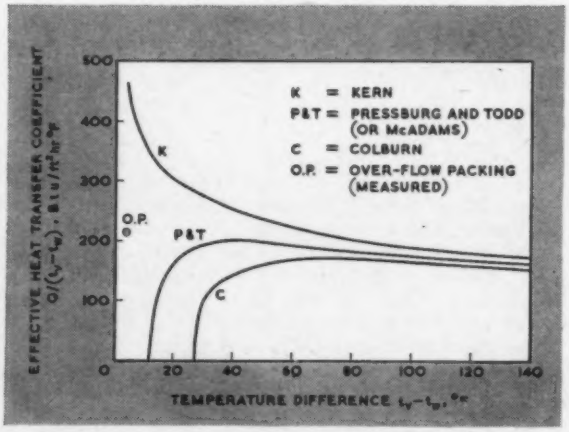


Fig. 2. Prediction of the heat-transfer coefficient for the condensation of a 71% methanol/29% n-propanol mixture by different methods.

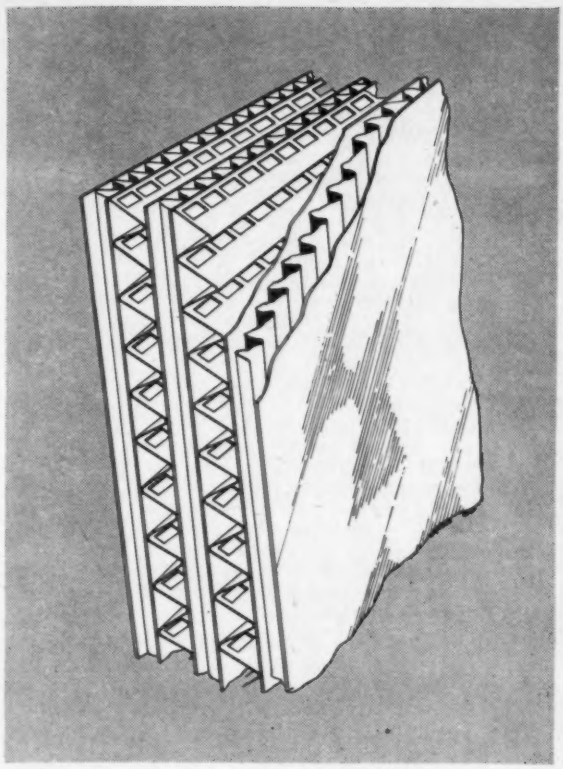


Fig. 3. An efficient condensing device—the overflow packing. (Courtesy: Marston Excelsior.)

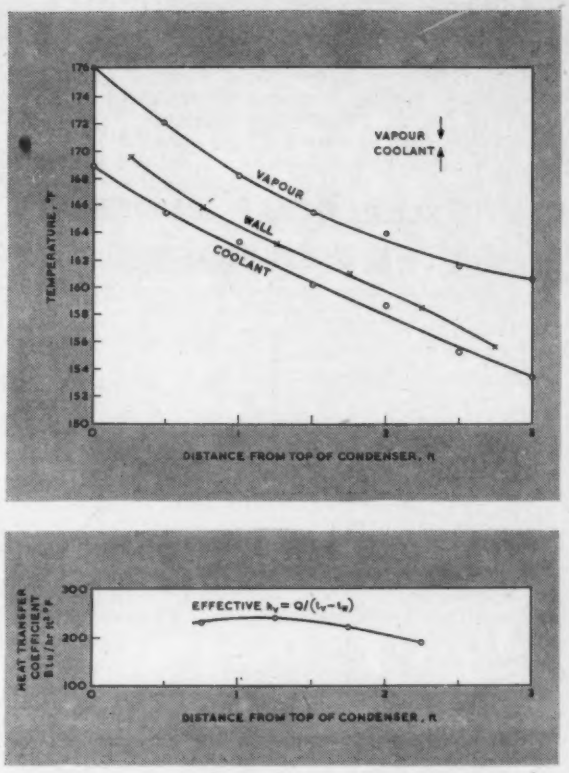


Fig. 4. Condensation of 71% menthanol—29% n-propanol vapour in overflow packing.

height of the condensing surface and to encourage turbulence in the condensate.

To promote mass transfer in and between the phases it is necessary to have the maximum degree of mixing in each phase and a large area of contact between them. It is also vital that the vapour and condensate should flow through the condenser together at a steady rate. Mixing in the vapour phase is readily achieved by employing a sufficient velocity to give turbulent flow, and it can be further enhanced by introducing changes of direction or area in the flow pattern. Liquid mixing is more difficult to achieve. It is aided by rippling and turbulence in the condensate film. However, it appears that if reliance is placed on the turbulence present in a film of condensate draining down the inside of a vertical tube under the influence of vapour drag, then a very long tube length is generally required to give a close approach to phase equilibrium. In many cases it would appear preferable to use more deliberate means for inducing mixing in the condensate and for bringing the vapour and condensate together.

One form of heat exchanger which embodies the required features employs a secondary surface construction with a corrugated and perforated matrix brazed between flat metal sheets (Fig. 3). This form of corrugation, called "over-flow" packing, was developed by one of the authors for reflux condensation with vapour flowing vertically upwards. For the duty under consideration the vapour mixture is brought in at the top and flows down co-currently with the condensate.

The performance of over-flow packing for mixed vapour condensation has been tested experimentally for mixtures of ethanol/methanol and methanol/n-propanol over a small range of conditions, and typical results are illustrated in Fig. 4. The test unit consisted of a single channel of packing 4 in. wide by 0.44 in. thick by 3 ft long, bounded on both faces by channels containing a normal multi-entry fin packing through which the cooling water passed. The unit was made of aluminium and thermocouples were located at intervals in the plates separating the streams as well as in the streams themselves, so that the separate heat-transfer coefficients could be determined. The experiments cover amounts of condensation up to about 90% of the feed rate, but do not extend to total condensation.

The uniform change of coolant temperature shows that condensation was taking place evenly throughout the condenser, while the regular variation of vapour temperature indicates that there is a close approach to phase equilibrium. The average heat-transfer coefficient under these conditions was 220 Btu/sq. ft hr °F for an average temperature difference between the vapour and wall of approximately 4°F. A point representing these average values is included in Fig. 2. It is thus demonstrated that the achievement of high heat-transfer coefficients during equilibrium-type condensation is possible. Further experiments are now being undertaken to extend the information available both with overflow packing and other forms of heat-transfer surface.

REFERENCES

[1] Colburn, A. P. "Proceedings of the General Discussion on Heat Transfer", *Inst. of Mech. Eng.*, 1951, 1.
[2] Colburn, A. P., and Drew T. B. *Trans. A.I.Ch.E.*, 1937, 33, 197.
[3] Kern, D. Q. "Process Heat Transfer", McGraw Hill, New York, 1950.
[4] McAdams, W. H., "Heat Transmission", third edition, McGraw Hill, New York, 1954.
[5] Pressburg, B. S., and Todd, J. B. *A.I.Ch.E., J.* 1957, 3, 348.
[6] van Es, J. P., and Heertjes, P. M. *Chem. Eng. Sci.*, 1956, 5, 217.

66.045.1; 66.074.8 : 66.012.7

OVERALL ECONOMICS OF PLATE-FIN HEAT EXCHANGERS

How the optimum design of this form of compact exchanger may be approached

by W. H. DENTON, B.Sc., F.Inst.P.

IN an earlier paper[1] the many advantages of plate-fin heat exchangers for low-temperature gas separation plant were discussed and methods developed for calculating the dimensions of plate-fin units to satisfy given heat-transfer and pressure-drop requirements.

The purpose of this article is first to analyse, in a general manner, the overall economics of the heat exchangers in low-temperature gas separation plant; this involves the power costs for the refrigeration load and pressure drop they introduce into a process and their capital cost. Secondly, the purpose is to develop criteria for the optimum design of plate-fin heat exchangers for which the sum of the above costs is a minimum. In turn, this minimum cost is shown to be an appreciable part of the total production costs of separating either oxygen from air or deuterium from hydrogen, as examples. The same analysis can be applied to other types of heat exchangers with different performance characteristics and capital costs.

For British conditions, especially for large-scale production, the energy cost is a much higher proportion of the total production cost (energy plus capital, etc.) in the case of low-temperature processes than it is for normal or high-temperature processes. This is because the low-temperature processes require the energy as power (and not heat) and because of the high power consumption associated with thermodynamic irreversibilities in equipment at low temperatures. For example, the irreversible entropy increases associated with the heat transferred in a heat exchanger are inversely proportional to the square of the absolute temperature, so that for a heat exchanger operating between room temperature and (say) 80°K the energy cost is more than ten times greater[1] than that for a similar heat exchanger with the same temperature difference but conserving heat in a process operating above room temperature. Consequently, since energy costs can be estimated fairly closely, while capital and other costs are frequently uncertain, calculations of the overall economics are likely to be more useful for low-temperature processes. This applies especially to the heat exchangers involved, since they are the main source of thermodynamic irreversibilities (the distillation step itself being potentially reversible) while, from the point of view of capital cost, they are a fairly large part of a gas separation plant operating below about 100°K.

In general, to perform an operation (namely, gas separation by distillation) at a low temperature T_1, while having to do all the necessary work in the form of gas compression at a temperature slightly above that of the surroundings (T_o) so that it can be rejected as heat to the surroundings, the feed, product and reflux gas streams have to be circulated counter-currently through heat exchangers between the temperatures T_o and T_1. The sole function of the heat exchangers is to reduce the resulting flow of heat from T_o to T_1 to an acceptable value, since additional work is required to absorb this heat at T_1 and reject it at T_o, i.e., provide refrigeration to maintain the heat balance over the whole system. By reducing the temperature differences in the heat exchangers the work of refrigeration is reduced proportionally, but the work to overcome pressure drop and the volume (capital cost) of the exchangers are both increased, so that there is an optimum heat exchanger design which will provide a minimum total cost for the heat-exchange step. Thus the overall economics of the heat exchangers can, in principle, be considered independently of all other thermodynamic irreversibilities in the process and independently of the distillation step itself. Only the pressures and the flows of the different gas streams are determined by the distillation step (i.e., the compression of feed and reflux gas streams) and by the means used to provide refrigeration. It is assumed here that the refrigeration is provided by the most thermodynamically efficient method, by doing external work in expansion engines at T_1 in one or more of the gas streams, when the pressures required are low enough to allow plate-fin units to be used.

The three cost factors contributing to the total cost of the heat exchange step in a process, the sum of which have to be minimised, are:

(1) power consumption for refrigeration;
(2) power consumption to overcome pressure drop; and
(3) capital cost of heat exchangers.

Total Power Consumption of Heat Exchangers

The total power consumption contributed by (1) and (2) is given by the sum of the irreversible entropy increases $(\Delta S_{irr})_{\Delta T}$ and $(\Delta S_{irr})_{\Delta P}$, respectively, due to the transfer of heat across temperature differences and due to the pressure drop in the heat exchanger. The corresponding work per gm-mole of gas is given by:

$$W = (W_{\Delta T} + W_{\Delta P}) = T_o \cdot M\left\{(\Delta S_{irr})_{\Delta T} + (\Delta S_{irr})_{\Delta P}\right\} \quad \ldots (21)$$

Assuming that the total heats, and therefore the tem-

Mr. Denton is with the Atomic Energy Research Establishment, Harwell.

Symbols used with this Paper

Q = heat, cals;
T_o = room temperature, °K;
T_1, T_2 = low temperatures, °K;
T_b = bulk gas temperature, °K;
T_s = metal surface temperature, °K;
$\Delta t = (T_b - T_s)$ = temperature difference, °K;
d_e = equivalent hydraulic diameter, cm;
A_1 = primary surface area per unit free volume, cm^{-1};
A_2 = secondary surface area per unit free volume, cm^{-1};
h_T = average surface heat-transfer coefficient at temperature, T, cals. cm^{-2} $°K^{-1}$ $sec.^{-1}$;
P = pressure of gas, gm. cm^{-2}
P_o = atmospheric pressure;
P_1, P_2 = average gas stream pressures;
L = length of heat exchanger, cm;
St = STANTON number, dimensionless;
f = friction factor, dimensionless;
Re = REYNOLDS number, dimensionless;
Pr = PRANDTL number, dimensionless;
A, B, a, b = constants for heat exchanger performance, dimensionless;
G = mass flow rate per unit free cross-sectional area, gm cm^{-2} $sec.^{-1}$;
F_m = mass flow rate;
F_M = molar flow rate;
g = acceleration of gravity, cm $sec.^{-2}$;
ρ = density of gas, gm. cm^{-3};
μ = viscosity of gas, gm cm^{-1} $sec.^{-1}$;
k = thermal conductivity of gas, cals. cm^{-1} $sec.^{-1}$ $°K^{-1}$;
C_p = specific heat of gas at constant pressure, cals. gm^{-1} $°K^{-1}$;
η_f = fin efficiency for heat transfer, dimensionless;
k_m = thermal conductivity of metal, cals. cm^{-1} $sec.^{-1}$ $°K^{-1}$;
t = thickness of fins, cm;
w_f = width of fins, cm;
ΔP = overall pressure drop, gm cm^{-2};
$\Delta P_1, \Delta P_2$ = pressure drops in gas streams, gm cm^{-2};
S = free cross-sectional area of heat exchanger per unit mass flow rate, cm^2 gm^{-1} $sec.$;
S_M = free cross-sectional area of heat exchanger per unit molar flow rate, cm^2 $gm\text{-}mole^{-1}$ $sec.$;
J = heat-transfer function, cm gm sec. units;
F = pressure-drop function, cm gm sec. units;
F_o = F at atmospheric pressure;
v = gas velocity, cm $sec.^{-1}$;
N = number of "velocity heads";
$(\Delta S_{irr})_{\Delta T}$ = irreversible entropy increase due to heat transfer, cals. gm^{-1} $°K^{-1}$;
$(\Delta S_{irr})_{\Delta P}$ = irreversible entropy increase due to pressure drop, cals. gm^{-1} $°K^{-1}$;
W = total work per gm-mole of gas, ergs. $gm\text{-}mole^{-1}$;
$W_{\Delta T}$ = work of refrigeration per gm-mole of gas, ergs. $gm\text{-}mole^{-1}$;
$W_{\Delta P}$ = work of pressure drop per gm-mole of gas, ergs. $gm\text{-}mole^{-1}$;
R = gas constant per gm-mole, ergs. $gm\text{-}mole^{-1}$ $°K^{-1}$;
M = molecular weight;
η_c = isothermal efficiency of compressor, dimensionless;
η_e = isentropic efficiency of expander, dimensionless;
φ = unbalancing flow fraction, dimensionless;
ΔT_o = warm end (T_o) bulk gas temperature difference, °K;
Cp_o = warm end (T_o) specific heat of gas, cals. gm^{-1} $°K^{-1}$;
c_1 = cost factor for power cost plus compressor capital cost, $£_{erg}^{-1}$;
c_2 = cost factor for heat exchanger capital cost, £, cm^{-2} $sec.^{-1}$;
£ = total cost of heat exchange step per gm-mole of gas, £, $gm\text{-}mole^{-1}$;
$£_{\Delta T}$ = cost of power for refrigeration for heat exchange step per gm-mole of gas, £, $gm\text{-}mole^{-1}$;
$£_{\Delta P}$ = cost of power for pressure drop for heat exchange step per gm-mole of gas, £, $gm\text{-}mole^{-1}$;
$£_c$ = capital charges for heat exchange step per gm-mole of gas, £, $gm\text{-}mole^{-1}$;
m = mechanical equivalent of heat, ergs. cal^{-1};
V = free volume of heat exchanger per unit molar flow rate, cm^3 $gm\text{-}mole^{-1}$ sec.;
$C_{(a,b)}$ = constant, a function of constants a and b; and
α, β and γ = constants relating costs to heat exchanger design parameters.

perature differences, are ideally independent of pressure,* the entropy increases in each heat exchanger can be calculated independently of the remainder of the thermodynamic cycle of the process and of the pressures used.

Balanced Heat Exchangers

First, for a balanced exchanger (two counter-current gas streams of equal heat capacities), if ΔT is the difference between the bulk temperatures of the two streams at a temperature T, then, integrating along the exchanger, per gm-mole of gas:

$$W_{\Delta T} = T_o \,.\, M\,(\Delta S_{irr})_{\Delta T} = T_o \,.\, M \int_{T_1}^{T_o} C_p \left(\frac{1}{T} - \frac{1}{T+\Delta T}\right) . \, dT$$

$$= Q\left(\frac{T_o}{T_1} - 1\right) \qquad (\Delta T \ll T)\dagger$$

where $Q = (M \,.\, C_p \,.\, \Delta T)$.

Thus $W_{\Delta T}$ is the minimum work of refrigeration, where Q is the net heat flow from T_o to T_1 and is constant along the exchanger, even if the specific heat of the gas, C_p, varies with temperature. M = molecular weight of the gas.

Also, assuming for the present that the average pressures of the two gas streams are the same,

$$W_{\Delta P} = T_o \,.\, M \,.\, (\Delta S_{irr})_{\Delta P}$$

$$= -(R \,.\, T_o)\left(\frac{\Delta P}{P}\right) \qquad (\Delta P \ll P)\dagger$$

where ΔP is the total pressure change in the heat exchanger in *both* gas streams and is negative, and P is the average pressure.

Finally, including the irreversible entropy increases in the associated compressor at T_o and expansion engine at T_1, the total work per gm-mole of gas is given by:

$$W = (W_{\Delta T} + W_{\Delta P}) = Q \,.\, \left(\frac{T_o}{T_1 \,.\, \eta_c \,.\, \eta_e} - 1\right) - \left(\frac{R \,.\, T_o}{\eta_c}\right)\left(\frac{\Delta P}{P}\right) \quad \ldots . (22)$$

* Such deviations from ideality only occur at the lowest temperatures, causing the local temperature differences to increase, but the overall effect is negligible for the low pressure ratios considered here.

where η_c and η_e are the isothermal and isentropic efficiencies of the compressor and expansion engine respectively.

Unbalanced Heat Exchangers

For an unbalanced exchanger in which the ratio of the warm to cold gas stream molar flow rates is $(1 + \varphi) : 1$, the total work is given by Equation (21) summed for both the closed cycle flow and the unbalancing flow, φ, plus the minimum reversible work, $\varphi \,.\, M \,.\, (\Delta H_\phi - T_o \,.\, \Delta S_\phi)$, required to take the unbalancing flow from the thermodynamic state it entered the exchanger to that which it left it, i.e., to cool it from T_o to T_1. By integrating along the exchanger, this is given, per gm-mole of the cold stream, by:

$$W = (W_{\Delta T} + W_{\Delta P}) = (Q - \varphi \,.\, M \,.\, \Delta H_\phi)\left(\frac{T_o}{T_1 \,.\, \eta_c \,.\, \eta_e} - 1\right) - \left(\frac{R \,.\, T_o}{\eta_c}\right)\left(\frac{\Delta P}{P} + \varphi \,.\, \frac{\Delta P_\phi}{P}\right) \quad \ldots.(23)$$

where ΔH_ϕ and ΔP_ϕ are the total heat and pressure changes, respectively, of the unbalancing flow and are both negative and Q is now based on the warm-end temperature difference, ΔT_o, i.e., $Q = (M \,.\, C_{po} \,.\, \Delta T_o)$ where C_{po} is the specific heat at T_o. Thus, the effect of the unbalancing flow is to increase the refrigeration load at T_1 by the amount of heat extracted from it and to add its own pressure-drop term.

Summing over all the Heat Exchangers in a Plant

In gas separation plant the unbalancing flows are usually for φ of the order of only 1%; e.g., to reduce temperature differences for the evaporation of solid impurities. Also, all the feed and product streams usually enter and leave at room temperature, so that, summed over all the exchangers in the plant, the sum of the unbalance flows in one direction ($\varphi + ve$) is exactly cancelled by the sum of those in the opposite direction ($\varphi - ve$). Summing Equation (23) over all the exchangers in such a plant and adding the necessary work of refrigeration to maintain the heat balance over all the unbalance streams since $\Sigma(\varphi \,.\, M \,.\, \Delta H\varphi)$ is not zero, but, neglecting very small second-order terms, the total power consumption contributed by all the heat exchangers in a plant is given by*:

$$\Sigma(F_M \,.\, W) = \Sigma F_M \,.\, (W_{\Delta T} + W_{\Delta P}) = \Sigma F_M \,.\, \left\{Q \,.\, \left(\frac{T_o}{T_1 \,.\, \eta_c \,.\, \eta_e} - 1\right) - \left(\frac{R \,.\, T_o}{\eta_c}\right)\left(\frac{\Delta P}{P}\right)\right\} \quad \ldots.(24)$$

where $Q = (M \,.\, C_{po} \,.\, \Delta T_o)$ and F_M is the cold stream molar flow rate. This is the same as that given by summing Equation (22) as if all the exchangers were balanced. We thus have the result that the total power consumption contributed by all the heat exchangers is independent of any unbalancing of the flows. The same result applies to a plant producing a liquid product ($\varphi + ve$), except for the additional work of refrigeration to remove all the necessary heat from the product at T_1, part of which is the minimum reversible work for liquefying the product.

In a plant with room temperature feed and product streams, when determining the optimum warm-end temperature differences, $(\Delta T_o)_{opt.}$, to balance the above power consumption against capital cost, the effect of unbalancing flows is that $(\Delta T_o)_{opt.}$ is reduced when φ is $+ve$, and vice-versa. The log. mean temperature difference in each heat exchanger is then made equal to $(\Delta T_o)_{opt.}$ for the balanced-flow case and the same optimum cross-section is used as for the balanced-flow case. It can then be shown that $\Sigma(F_M \,.\, Q)$ and $\Sigma\left(F_M \,.\, \frac{\Delta P}{P}\right)$ in Equation (24), and hence the total power consumption which is the main part of the total cost, are practically independent of unbalancing, although the corresponding total volume of all the heat exchangers is increased due to unbalancing. Therefore, the following treatment of the overall economics is confined to the balanced-flow case, and when applying the results to optimise the design of an unbalanced exchanger the same optimum cross-section as for balanced flows is used and the optimum log. mean temperature difference is made equal to $(\Delta T_o)_{opt.}$ for balanced flows.

Total Cost (Power plus Capital) of the Heat-Exchange Step and Optimisation of the Design Parameters

The minimised total cost incurred by the heat exchangers in a gas separation plant can be calculated for the case of plate-fin units, using Equations (11) and (12) given in the earlier paper[1] for the dimensions (capital cost) of the units and Equation (24) for the total power consumption they contribute to the process. It is first necessary to calculate the optimum design parameters for which the total cost is a minimum.

A two-stream balanced-flow counter-current unit is considered, operating between temperatures T_o and T_1, where the average pressures of the two gas streams are P_2 and P_1, respectively, and $P_2 > P_1$. The different pressures only effect the pressure drops which are inversely proportional to pressure. If the two gas stream channels are considered separately, the optimum length is greater and the optimum cross-section is less for the higher pressure stream, but, in practice, their lengths have to be the same. Even if the cross-section of the higher pressure stream is reduced (by reducing alternate plate spacings), a negligibly small reduction in the total cost results, since the reduction in heat exchanger volume is offset by an increase in the temperature difference in the high-pressure stream (see Equation (3), Ref. 1) and therefore in the power for refrigeration. A symmetrical design of heat exchanger is therefore assumed, so that the cross-section and temperature difference are

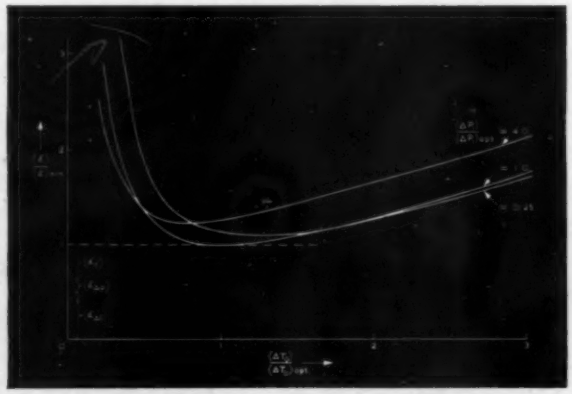


Fig. 1. Optimisation of temperature difference (ΔT_o) *and pressure drop* (ΔP_1) *for minimum total heat exchanger costs* (£). (a = 0.24, b = 0.31.)

† It can be seen in Table I (p. 48) that $\Delta T \ll T$ and $\Delta P \ll P$ for the practical cases.

* Equations (22), (23) and (24) apply to all types of heat exchangers even if the gas streams are different gases or mixtures of gases, provided their molar specific heats are the same, e.g., most diatomic gases, such as O_2, N_2 and H_2.

the same for both gas streams. The total pressure drop term, $\left(\frac{\Delta P}{P}\right)$, in Equation (24) has to be modified and is given by:

$$\left(\frac{\Delta P}{P}\right)_{1,2} = \left(\frac{\Delta P_1}{P_1} + \frac{\Delta P_2}{P_2}\right) = \left(\frac{\Delta P_1}{P_1}\right) \cdot \left(1 + \frac{P_1^2}{P_2^2}\right) \quad \ldots.(25)$$

The temperature difference term, (Δt), in Equations (11) and (12) has also to be modified to take account of variations of the specific heat of the gas with temperature (e.g., for hydrogen) when, for the present case of balanced flows, (Δt) for both gas streams is given by:

$$(\Delta t)_T = \tfrac{1}{2}\left(\frac{C_{po}}{C_{pT}}\right) \cdot (\Delta T_o) \quad \ldots.(26)$$

The two independent design parameters to be optimised for minimum total cost are:

(1) the warm-end temperature difference, ΔT_o; and

(2) the pressure drops in the two gas streams.

The total cost contributed by each such heat exchanger to a low-temperature process is then given by:

Total cost per gm-mole of gas, $\pounds = (\pounds_{\Delta T} + \pounds_{\Delta P} + \pounds_C)$, the sum of the costs of work of refrigeration ($\pounds_{\Delta T}$), of work to overcome pressure drop ($\pounds_{\Delta P}$), and of capital charge ($\pounds_C$) for the heat exchanger itself.

From Equations (24), (25), (26), (11)[1] and (12)[1],

$$\left.\begin{aligned} \pounds_{\Delta T} &= c_1 \cdot W_{\Delta T} = \alpha \cdot (\Delta T_o), \\ \pounds_{\Delta P} &= c_1 \cdot W_{\Delta P} = \beta \cdot (S_M)^{-(2+a-b)} \cdot (\Delta T_o)^{-1} \\ \text{and } \pounds_C &= c_2 \cdot V = \gamma \cdot (S_M)^{(1-a)} \cdot (\Delta T_o)^{-1} \end{aligned}\right\} \quad \ldots.(27)$$

where S_M is the free cross-sectional area per unit molar flow rate, so that:

$$S_M \cdot = M \cdot S.$$

c_1 is a constant accounting for the cost of power plus an appropriate capital charge for the gas compressor and c_2 is a constant accounting for the overall capital cost of the heat exchanger per unit *free* volume of the block. V is the free volume required for both gas streams per unit molar flow rate, so that

$$V = 2 \cdot L \cdot S_M$$

where L is the length of the exchanger block.

The constants α, β and γ are given by Equations (24), (25), (26), (11) and (12). The pressure-drop function F (which is inversely proportional to pressure) is replaced by $\left(\frac{P_o}{P_1}\right) \cdot F_o$ in Equation (11) for (ΔP_1), where F_o is the value of F at atmospheric pressure, P_o.

Thus:

$$\alpha = c_1 \cdot m \cdot M \cdot C_{po} \cdot \left(\frac{T_o}{T_1 \cdot \eta_c \cdot \eta_e} - 1\right)$$

$$\beta = c_1 \cdot (M)^{(2+a-b)} \cdot \frac{R \cdot T_o \cdot (de)^{(a-b)}}{2\,\eta_c \cdot C_{po}} \left(\frac{P_o}{P_1^2}\right) \left(1 + \frac{P_1^2}{P_2^2}\right) \int_{T_1}^{T_o} \left(\frac{C_p^2 \cdot F_o}{J}\right) \cdot dT$$

$$\text{and } \gamma = c_2 \cdot (M)^a \cdot \frac{(d_e)^{(1+a)}}{C_{po}} \cdot \int_{T_1}^{T_o} \left(\frac{C_p^2}{J}\right) \cdot dT \quad \ldots.(28)$$

where m is the mechanical equivalent of heat. Fig. 1 is a plot of the sum of Equations (27) with (S_M) expressed in terms of (ΔP_1) and (ΔT_o) from Equations (11) and (25). It shows that there are optimum values for both (ΔP_1) and (ΔT_o) which give a minimum total cost $(\pounds)_{Min}$. This cost is much more sensitive to variations of (ΔT_o) than of (ΔP_1) and it increases rapidly if (ΔT_o) is reduced below its optimum value. By partially differentiating $(\pounds)$ by (ΔT_o) and (S_M), respectively, these optimum values are given by

$$(\Delta T_o)_{opt.} = \alpha^{-\frac{1}{2}} \cdot \beta^{\frac{1}{2}\left(\frac{1-a}{3-b}\right)} \gamma^{\left(\frac{2+a-b}{2(3-b)}\right)} \cdot C_{(a,b)} \quad \ldots.(29)$$

$$\text{and} \quad (S_M)_{opt.} = \left\{\frac{\beta}{\gamma} \cdot \left(\frac{2+a-b}{1-a}\right)\right\}^{\frac{1}{(3-b)}} \quad \ldots.(30)$$

$$\text{or, from (11):} \quad \left(\frac{\Delta P}{P}\right)_{1,2\,opt.} = \left(\frac{\Delta P_1}{P_1}\right)_{opt.} \left(1 + \frac{P_1^2}{P_2^2}\right)$$

$$= \alpha^{\frac{1}{2}} \cdot \beta^{\frac{1}{2}\left(\frac{1-a}{3-b}\right)} \gamma^{\left(\frac{2+a-b}{2(3-b)}\right)} \left(\frac{1-a}{3-b}\right) \left(\frac{\eta_c}{c_1 \cdot R \cdot T_o}\right) \cdot C_{(a,b)} \quad \ldots.(31)$$

where the constant, $C_{(a,b)} =$

$$\left\{\frac{(3-b)^{(3-b)}}{(1-a)^{(1-a)} \cdot (2+a-b)^{(2+a-b)}}\right\}^{\frac{1}{2(3-b)}}$$

The corresponding length of the exchanger, from Equation (12), is:

$$(L)_{opt.} = \alpha^{\frac{1}{2}} \cdot \beta^{-\frac{1}{2}\left(\frac{1+a}{3-b}\right)} \cdot \gamma^{\left(\frac{4+a-b}{2(3-b)}\right)} \left(\frac{1-a}{2+a-b}\right)^{\frac{1}{(3-b)}} \left(\frac{2+a-b}{3-b}\right) \left(\frac{1}{c_2}\right) C_{(a,b)} \quad \ldots.(32)$$

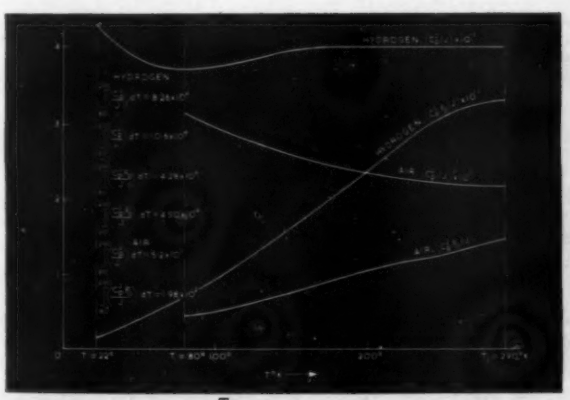


Fig. 2. Evaluation of $\int_{T_1}^{T_o} \left(\frac{C_p^2}{J}\right) \cdot dT$ *and* $T_o T_1 \left(\frac{C_p^2 F_o}{J}\right) \cdot dT$ *for hydrogen and air (cgs units).*

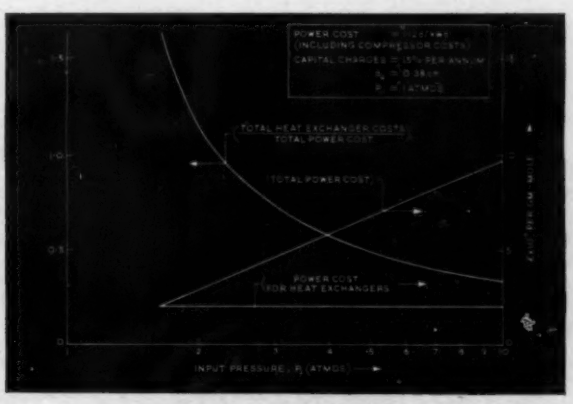


Fig. 3. Optimised total heat exchanger costs relative to total power cost for plate-fin units for air (290°-80°K) or hydrogen (290°-22°K).

Finally, the minimum total cost per gm-mole of gas is:

$$(£)_{Min.} = 2 \cdot (£_{\Delta T})_{Min.} = \frac{2(3-b)}{(1-a)} (£_{\Delta P})_{Min.} = \frac{2(3-b)}{(2+a-b)} (£_c)_{Min.}$$

$$= 2 \cdot \alpha^{\frac{1}{3}} \cdot \beta^{\frac{1}{3}\left(\frac{1-a}{3-b}\right)} \gamma^{\frac{(2+a-b)}{2(3-b)}} C_{(a,b)} \quad \ldots.(33)$$

We thus have the interesting result, whatever the temperature range, the pressure or the physical properties of the gas, that when the total cost is a minimum, then the refrigeration power cost, the pressure drop power cost and the capital cost, respectively—whatever their actual values—are always in the proportions:

$$(3-b):(1-a):(2+a-b)$$

or, for typical values of a and b, approximately:

$$3:1:2.$$

Also, whatever the flow régime (a and b values), half the minimum total cost is power for refrigeration.

The Viscous Flow Region

It should be noted that as viscous flow conditions are approached (a and b tend to unity), although $(S_M)_{opt.}$ would appear to tend to infinity and $(L)_{opt.}$ and the pressure drop tend to zero, there are practical restrictions which prevent this and make it necessary to modify the above analysis. These restrictions are due principally to longitudinal heat conduction along the heat exchanger block itself and gas distribution problems. They do not arise for the flow régimes ($a = 0.24$ and $b = 0.31$) considered here. (See Table I.)

Dependence of $(£)_{Min.}$ on Cost of Electric Power, Capital Cost, Plate-fin Geometry and Flow Régime

For practical a and b values, about two-thirds of the minimum total cost is for power and only about one-third is for capital charges. Thus, as is shown directly by Equation (33), $(£)_{Min}$ varies as the cost of electric power raised to the power 2/3 and varies as the capital cost raised to the power 1/3. These proportions should not be very different when the additional pressure drops in a plant and the additional capital costs for thermal insulation, etc., are taken into account.

The dependence of $(£)_{Min}$ on the plate and fin geometry is mainly that of a reduction in block volume as the plate and fin spacings are reduced, which is more rapid as the viscous flow region is approached (as a and b tend to unity), as discussed in Ref. 1. The effect on the overall economics can be expressed, from Equations (28) and (33), as:

$$(£)_{Min.} \propto \left(\frac{F_o}{J}\right)^{0.14} \times (c_2)^{0.37} \times (de)^{0.46} \quad \ldots.(34)$$

for typical REYNOLDS numbers ($a = 0.24$, $b = 0.31$). Since d_e is inversely proportional to the surface area per unit *free* volume, there would appear to be an appreciable overall saving to be gained if the plate and fin spacings could be reduced. This would be offset first by an increase in the volume of metal and hence in the capital cost constant, c_2, which has to take account of the fact that the capital cost is based on *total* volume. The additional metal volume is about 20% of the free volume for the present plate and fin spacings. Secondly, for very close fin spacings, as the spacings are reduced the ratio of the pressure drop to heat transferred increases fairly rapidly[2] due to increased inertia losses, so that the factor $\left(\frac{F_o}{J}\right)$ increases proportionally. There is thus a fairly sharp limit to the savings to be obtained by reducing the plate and fin spacings.

As the viscous flow region is approached these inertia losses are less and also the above exponents for $\left(\frac{F_o}{J}\right)$, (c_2) and (d_e) tend towards 0, ½ and 1 respectively (with practical restrictions), representing a much more rapid reduction in $(£)_{Min.}$ as the spacings are reduced. However, it appears from this analysis that the REYNOLDS numbers for the optimum design in low-temperature plant are far too high for these advantages to be exploited, as shown in Table I. Also, there are obviously difficult problems and extra expense in manufacturing closer plate and fin spacings.

Dependence of $£_{Min.}$ and Optimum Design Parameters on Physical Properties of Gas, Temperature Range and Pressures

Equations (28) and Equations (7a) and (8a) in Ref. 1 show that the only physical properties of the gas which are influential are its density and its molar specific heat at normal temperature $(M \cdot C_{po})$. The very small effects of variations of specific heat with temperature and the very small contribution of viscosity—as $(\mu)^{0.1}$—can both be neglected, while the density can be assumed approximately proportional to the molecular weight, M. The approximate dependences on the temperature range and pressures can also be expressed simply, but not so accurately, since in Equations (28), when $T_1 < 100°K$, the integral for β is approximately proportional to $(T_o^2 - T_1^2)$ and that for γ is approximately proportional to $(T_o - T_1)$ (see Fig. 2). Thus, the approximate overall dependences on physical properties, temperature range and pressures are all given, from Equations (28) to (33), for $T_1 < 100°K$, by:

$$(£)_{Min.}, (V)_{opt.} \text{ and } \left(\frac{\Delta P}{P}\right)_{1,2\,opt.} \propto \frac{(T_o - T_1)}{(T_1)^{0.5}} \cdot (M)^{\frac{(1+2a-b)}{2(3-b)}} \cdot (M \cdot C_{po})^{0.5} \cdot \left[\frac{1}{P_1^2}\left(1 + \frac{P_1^2}{P_2^2}\right)\right]^{\frac{1}{3}\left(\frac{(1-a)}{(3-b)}\right)}$$

$$(\Delta T_o)_{opt.} \propto (T_1)^{0.5} \cdot M^{\frac{(1+2a-b)}{2(3-b)}} \cdot (M \cdot C_{po})^{-0.5} \cdot \left[\frac{1}{P_1^2}\left(1 + \frac{P_1^2}{P_2^2}\right)\right]^{\frac{1}{3}\left(\frac{1-a}{3-b}\right)}$$

$$\text{and } (S_M)_{opt.} \propto (M)^{\left(\frac{1-b}{3-b}\right)} \cdot \left[\frac{1}{P_1^2}\left(1 + \frac{P_1^2}{P_2^2}\right)\right]^{\frac{1}{(3-b)}} \quad \ldots.(35)$$

These relationships show that $(£)_{Min.}$ is very sensitive to the lower absolute temperature, T_1—varying as $\frac{(T_o - T_1)}{(T_1)^{0.5}}$ for all flow régimes—that it is significantly dependent on the pressures for typical REYNOLDS numbers, especially on the lower pressure, P_1—varying as $\frac{1}{(P_1)^{0.3}} \times \left(1 + \frac{P_1^2}{P_2^2}\right)^{0.15}$ for $a = 0.24$, $b = 0.31$—but that it depends surprisingly little on the physical properties of gases with the same molar specific heats (e.g., diatomic gases such as H_2, N_2 and O_2), when $(£)_{Min.}$ only varies as $(M)^{0.2}$ for $a = 0.24$, $b = 0.31$, due to the effect of density on the pressure drop. The dependence of $(£)_{Min.}$ on the molar specific heat is as $(M \cdot C_{po})^{0.5}$ for all flow régimes, since this only affects the refrigeration load.

The optimum temperature difference and the optimum pressure drop are similarly sensitive to T_1 and have exactly the same dependences on pressure and physical properties, apart from the molar specific heat term.

The optimum cross-section is independent of T_1 and is much more sensitive to the pressures. This confirms the earlier result[1] that no significant reduction in volume of the exchanger can be obtained by varying its cross-section optimally with temperature.

When your tanks are lined with Ensecote, you can stop worrying about the condition of the contents. Ensecote protective linings will protect almost anything you can put inside a tank, *any* size tank. They are without taste or odour, they are easy to clean, and sterilize—and they are economical to apply. Ensecote protective linings add years to a tank's life.

ENSECOTE *Lithcote* PLASTIC LININGS

For literature and technical advice write to: NEWTON CHAMBERS & CO. LTD., THORNCLIFFE, SHEFFIELD, ENGLAND

Agents in Belgium, France and Holland: FOURS LECOCQ ET ATELIERS DE TRAZEGNIES REUNIS, S.A., TRAZEGNIES, BELGIUM

Reduction of $(£)_{Min.}$ by Providing Refrigeration at Intermediate Temperature Levels

Since $(\Delta T_o)_{opt.}$ is proportional to $(T_1)^{0.5}$ a significant reduction in $(£)_{Min.}$ can be expected by reducing the temperature differences at the lower temperatures, especially for low values of T_1. This is done by providing part of the refrigeration at at least one intermediate temperature level, thereby reducing the temperature difference below this temperature level. For example, by expanding the warm gas stream in two expansion engines at T_1 and T_2, respectively, where T_2 is now the lowest temperature and $T_o > T_1 > T_2$.

The foregoing treatment of the overall economics from Equations (27) to (33) is then applied to the separate temperature ranges and their separate contributions to the total cost added together. (ΔT_o) and (C_{po}) are replaced by the appropriate values for the warm end of each range. The appropriate temperature limits are used for the integrals for β and γ in Equations (28) and, except for the highest temperature range, α is replaced by:

$$\alpha = \frac{c_1 \cdot m \cdot M \cdot C_{pT_1}}{\eta_c \cdot \eta_e} \left(\frac{T_o}{T_1}\right) \left(\frac{T_1}{T_2} - 1\right)$$

for the range T_2 to T_1, for example.

For the general case of N expansion engines between T_o and T_N, it can be shown, making the above approximations for temperature dependences, that the total cost is a minimum when both the absolute temperatures at which the engines are located and the temperature differences are in the same geometric progression. Thus, if $(T_n)_{opt.}$ is the optimum temperature of the n^{th} engine and $(\Delta T_n)_{opt.}$ is the optimum temperature difference below $(T_n)_{opt.}$,

$$\frac{(T_n)_{opt.}}{T_o} \simeq \frac{(\Delta T_n)_{opt.}}{(\Delta T_o)_{opt.}} \simeq \left(\frac{T_N}{T_o}\right)^{\frac{n}{N}} \qquad \ldots.(36)$$

For example, for cooling hydrogen to 22°K, with one intermediate refrigeration level at T_1,

$$T_o = 290°,\ T_N = 22° \text{ and } N = 2,$$

$$\text{giving } (T_1)_{opt.} \simeq 80°\text{K and } \frac{(\Delta T_1)_{opt.}}{(\Delta T_o)_{opt.}} \simeq 0.28$$

Substituting these conditions into the foregoing analysis, the total cost for the temperature range 290°-22°K is reduced below that for a single refrigeration level by a worth-while factor of 0.8, as shown in Table I.

Calculated Examples of $(£)_{Min.}$ and the Optimum Design Parameters

Table I gives examples of figures for $(£)_{Min.}$ calculated from the foregoing formulæ, for cooling both air and hydrogen to 80°K and for cooling hydrogen to 22°K (with one and with two refrigeration levels), together with the corresponding optimum temperature differences, pressure drops and exchanger dimensions. These figures are for average pressures of $P_1 = 1$ atm. and $P_2 = 5$ atm., which are fairly near to those required for the distillation of air or hydrogen.

These calculations are in c.g.s. units and £ sterling and are based on the following economic conditions.

(1) Electrical power at 1.0d. per kWh.
(2) Total capital charges at 15% per annum and 8000 hours' operation per annum.
(3) Gas compressor capital costs of £20 per horsepower, equivalent to an increase in the cost of electrical power of 0.12d. per kWh based on (2) above.
(4) Heat exchanger capital costs, including headers and directly associated manifolding of £100* per cu. ft of block volume (= 1.17 × *free* volume in this example).
(5) $\eta_c = \eta_e = 0.75$ for large-scale plants.

Thus: $c_1 = 1.30 \times 10^{-16}$ £.erg.$^{-1}$
and $c_2 = 0.216 \times 10^{-10}$ £.cm^{-3} sec.$^{-1}$.

A multi-entry-type plate-fin design is assumed with ten 0.006-in.-thick fins per inch, ¼ in. plate spacing and 0.030-in.-thick plates, when:

$$d_e = 0.38 \text{ cm.}$$

The J and F_o values used are those plotted in Fig. 6, of Ref. 1, for similar multi-entry fins for the REYNOLDS number range 700-10,000, the optimum REYNOLDS numbers in Table I lying well within this range. The integrals for Equations (28) are evaluated graphically in Fig. 2.

These optimum REYNOLDS numbers show that the advantage of the rapid reduction in $(£)_{Min.}$ as the plate and fin spacings are reduced, which sets in for viscous flow, cannot be exploited in low-temperature plant because of the higher REYNOLDS numbers associated with the higher densities and lower viscosities, even for these relatively close fin spacings.

Comparison Between Calculated and Practical Figures for Air Separation Plant

The calculated optimum temperature differences and pressure drops in Table I may well be modified in practice due to other economic considerations in a plant, bearing in mind the sensitivity of the total cost to these parameters as shown in Fig. 1. However, the calculated value for $(\Delta T_o)_{opt.} = 5.6°$K for cooling air to 80°K (which represents the main contribution to the total cost) is remarkably close to the values used in large air separation plant using plate-fin exchangers; it is about 50% larger than the values used in air separation plant using regenerators, due mainly to the lower capital cost of regenerators.

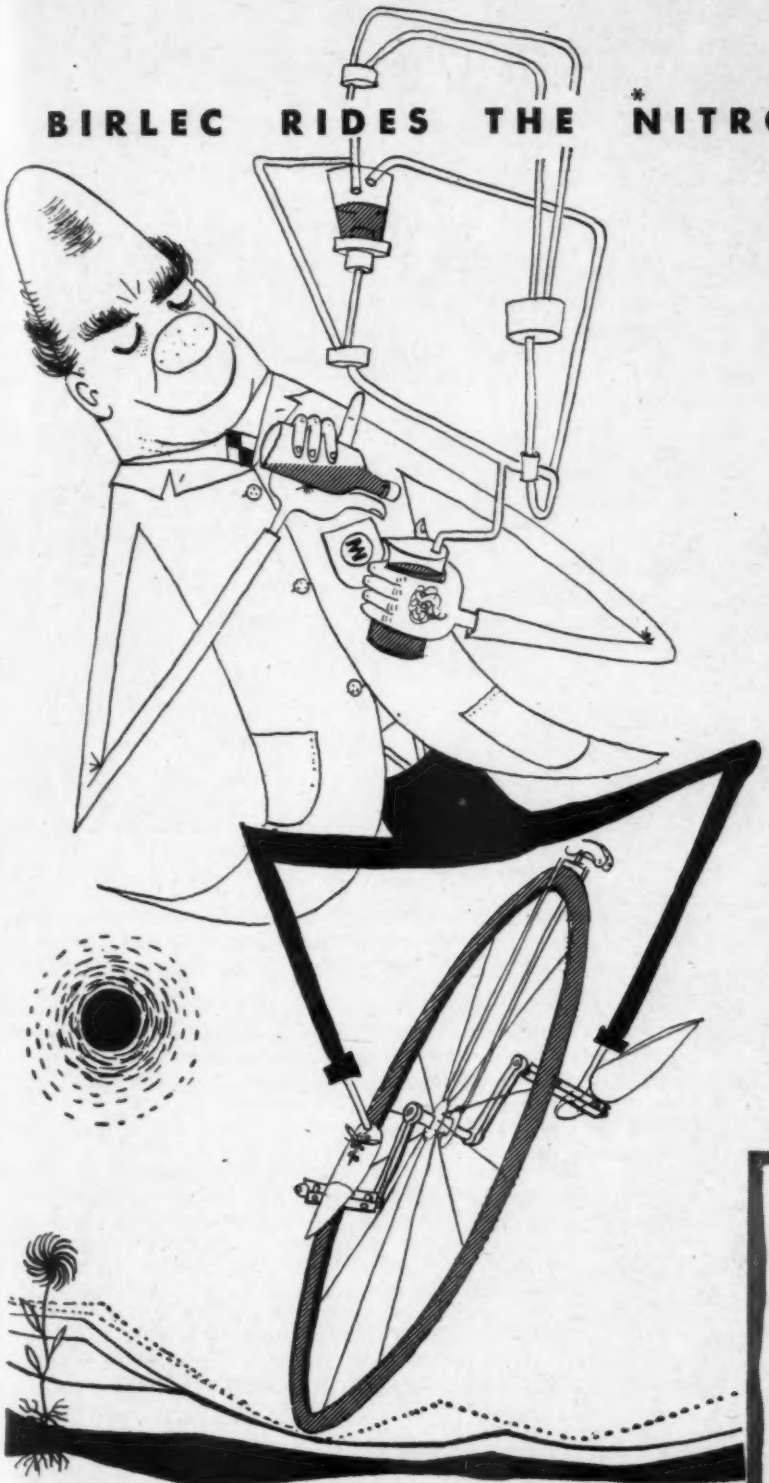
The corresponding calculated value for $\left(\frac{\Delta P_1}{P_1}\right)_{opt.} = 0.09$ is somewhat lower than (about one-third of) the overall pressure drop used in air separation plant, which is to be expected because it is the block pressure drop only which is only a part of the overall pressure drop in a plant.

Although there is this good agreement between the calculated optimum design parameters and those arrived at by practical experience, it should be stressed that the figures in Table I for $(£)_{Min.}$ are bound to be appreciably lower than the overall practical figures. They do not include either the additional (unavoidable) pressure-drop costs occurring in headers, manifolding, pipes and valves, or the additional capital costs of the associated pipes, valves, thermal insulation and structural work. Since these extra costs are not very dependent on the block dimensions, they should not alter the foregoing analysis, or significantly alter the resulting values for the optimum design parameters, but these extra costs have to be added to the calculated figures for $(£)_{Min.}$ Thus, the values for $(£)_{Min.}$ in Table I may well be increased by at least one-third, e.g., by increasing the pressure drop by a factor of 2 and the capital cost by a factor of 1.5, when $(£)_{Min.}$ is increased by 32% and the division between power and capital costs would remain almost the same.

A direct comparison can be made between the calculated *power* cost of the heat exchange step, $(£_{\Delta T} + £_{\Delta P})$ for cooling air to 80°K and that in a LINDE-FRANKL tonnage oxygen plant, for which the optimum breakdown of the total power consumption of the process is well established, although it applies to regenerators. The total power consumption of the heat exchangers (based on the overall pressure drop) in a LINDE-FRANKL plant is given[3] as 9% of the total power consumption for a compressor efficiency of 58% or, if the associated irreversibilities in

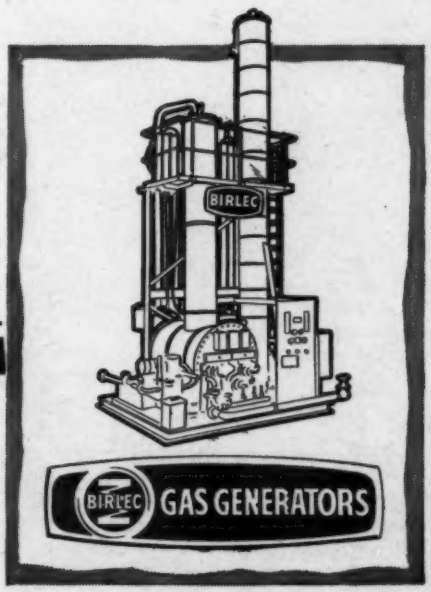
* This is an assumed round figure for the purpose of these calculations. In commercial practice it can, of course, vary with plant size, complexity of design, degree of standardisation, etc. The final total cost is not very sensitive to changes in this figure.

BIRLEC RIDES THE * NITROGEN CYCLE



We build nitrogen generators—and a wide range of other gas generators, too. The fact had better be stated quickly before the purists point out that the nitrogen cycle also concerns organic matter: no, we do not make nitrogen from decomposing animal tissue. Specially designed generators separate nitrogen from air by the combustion of fuel, which converts the oxygen content into carbon dioxide and water. We deal with these waste constituents, too, but one of our engineers will explain all that. Protective atmospheres for furnaces, including annealing of transformer laminations; purging of electric lamps and handling of semi-conductors; prevention of oxidation in chemical processes and in packing sensitive foodstuffs; handling of inflammable liquids in petroleum refining; all these processes employ nitrogen protection. Estimates of capital and running costs for Birlec nitrogen generators are available on request.

*. . . *if there were other gas cycles we'd ride those also, as we build generators for hydrogen, carbon dioxide, inert gases, and furnace atmospheres, in addition to nitrogen.*



Dryer and Gas Plant Division of
BIRLEC LIMITED
An A.E.I. Company
ERDINGTON · BIRMINGHAM 24 · Tel: EAST 1544

LONDON
SHEFFIELD
GLASGOW
NEWCASTLE-ON-TYNE
JOHANNESBURG

SM/B4995

TABLE I—Typical Minimum Total Costs and Optimum Design Parameters for Counter-current Heat Transfer in Plate-fin Units in the Temperature Ranges 290°-80°-22°K for Average Gas Stream Pressures of 1 and 5 atm., respectively, for Air and Hydrogen

Gas	*Air*	*Hydrogen*			
				290°–22°	
Temperature range	290°–80°	290°–80°	290°–22°	290°–80°	80°–22°
Refrigeration at	80°	80°	22°	80°	22°
$(£)_{Min.}$, £. gm-mole^{-1}	2.5×10^{-6}	1.4×10^{-6}	3.0×10^{-6}	1.4×10^{-6}	1.0×10^{-6} (Total $= 2.4\times10^{-6}$)
$(\Delta T_o)^*_{opt.}$, °K	5.6°	3.3°	1.8°	3.3°	0.9°
$\left(\frac{\Delta P_1}{P_1}\right)^\dagger_{opt.} \simeq \left(\frac{\Delta P}{P}\right)_{1,\,2\ opt.}$ dimensionless	0.09	0.05	0.10	0.05	0.04 (Total = 0.09)
$(S_M)_{opt.}$, cm^2 gm-mole^{-1} sec.	29.0	14.0	13.0	14.0	7.6
$(L)_{opt.}$, m.	13	15	35	15	20 (Total = 35)
$(R_e) \times 10^{-6}$	2–7	0.6–1.7	0.6–5	0.6–1.7	3–9

* For unbalanced exchangers, $(\Delta T_o)_{opt.}$ corresponds to the optimum log. mean temperature difference.

† (ΔP_1) is the contribution by the heat exchanger blocks *alone* to the overall pressure drop in the low-pressure stream, e.g., the excess pressure of the distillation column above atmospheric pressure.

the compressor are included (as in this analysis), as 15.5% of the total power consumption. Therefore, neglecting the very small proportion (4%) of the feed air in the very high pressure stream in the L.F. process, since the average pressures of the gas streams in the regenerators are $P_2 = 5.5$ atm.[3] and $P_1 \simeq 1$ atm.,

$$(£_{\Delta T} + £_{\Delta P})_{opt.} = c_1 \times 0.155 \times \frac{R\,.\,T_o\ ln\ (5.5)}{0.58}$$

$$= 1.4 \times 10^{-6}\ £.\text{gm-mole.}^{-1}$$

based on the same cost constant, c_1, used previously.

The corresponding calculated figure for plate-fin units, from Equation (33), after re-optimising for $\eta_c = 0.58$, and increasing the pressure drop by a factor of 2 to take account of the overall pressure drop (this increases the figure by 22%),

$$= 2.3 \times 10^{-6}\ £.\text{gm-mole.}^{-1}$$

This is higher than the practical figure for regenerators by about the factor to be expected when the plate-fin units are optimised for their much higher capital cost.

The Minimum Total Cost of the Heat-exchanger step expressed as a Percentage of the Total Power Cost for Low-temperature Gas Separation

Since the function of the heat exchangers in a process is to reduce the total power consumption, it is reasonable to express the total cost of the heat exchanger step (£) as a percentage of the total power cost.

In practically all these processes the product and reflux gas streams leave the plants at atmospheric pressure, so that the total power cost is that to compress the feed and reflux streams from atmospheric pressure, P_o, to higher input pressures, P_2, so that for each gas stream:

$$\text{Total power cost} = c_1 \frac{R.T_o}{\eta_c}\ ln\ (P_2/P_o) \text{ per gm-mole.}$$

where P_2 is determined mainly by the distillation step.

Since the pressure drop occurs mainly in the distillation section of the plant (across reflux expansion valves and across engines) the average pressures of the two gas streams in each exchanger are approx. atmospheric pressure and the higher pressure P_2, respectively. The corresponding minimum total cost of the heat-exchanger step, $(£)_{Min.}$, is then given by Equation (33) in terms of P_2. The ratio of $(£)_{Min.}$ to the total power cost for each gas stream then depends only on the input pressure, P_2, for a given gas and temperature range. This ratio is plotted against P_2 in Fig. 3 together with the total power cost and the power cost for the heat exchanger, after increasing the calculated values of $(£)_{Min.}$ by the proposed 32% to be closer to the overall practical figure. These curves apply to both air separation and hydrogen distillation (with two refrigeration levels) since $(£)_{Min.}$ happens to be nearly the same for both these cases. (See Table I.)

The power cost of the distillation step is not completely independent of the power cost of the heat exchange step, since there is some interdependence between them, depending on the thermodynamic cycle of the process, but the total power cost and the pressures are still determined mainly by the distillation step except for very low-pressure cycles. As the pressure is reduced, it can be seen in Fig. 3 (for plate-fin units) for the limiting case of the input pressure $P_2 = 1.65$ atm. that all the power consumption is required to overcome the irreversibilities in the heat exchanger and none is available for the distillation step. The fraction of the total power cost incurred by the heat exchangers then falls rapidly for higher values of P_2.

For air separation with a single feed gas stream at an input pressure of (say) 6.5 atm. for plate-fin units, the minimum total cost of the heat-exchange step is seen in Fig. 3 to be 42% of the total power cost which, in turn, is at least half of the total production cost (power plus capital, etc.).

For hydrogen distillation this percentage is even greater, since, because of the higher reflux ratio, an additional reflux gas stream of at least twice the feed stream is required and this is only compressed to an input pressure of considerably less than 6.5 atm.

REFERENCES

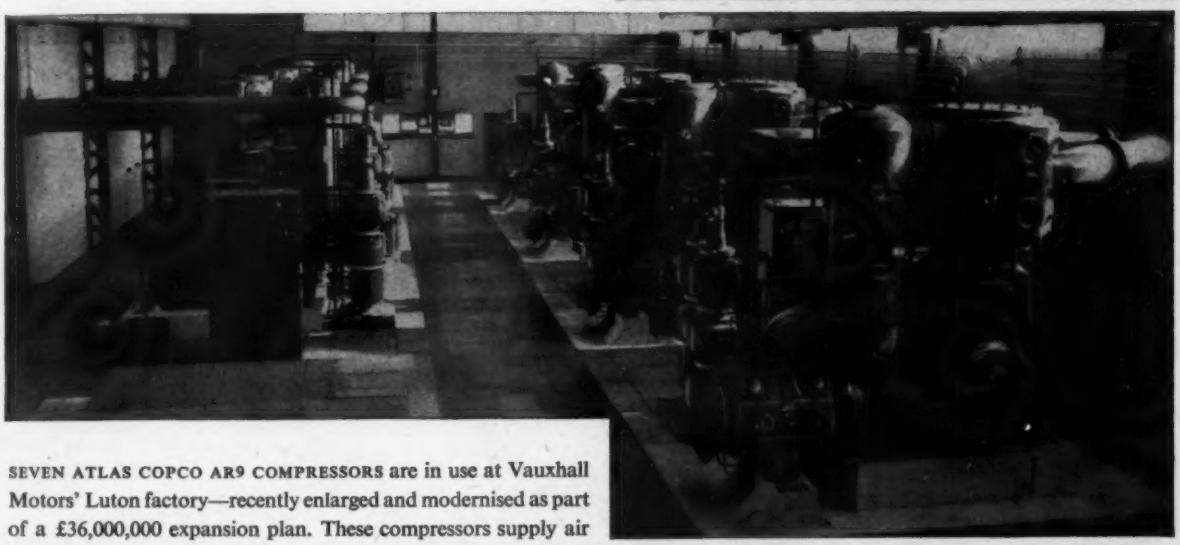
[1] Denton, W. H., and Ward, D. E. "Plate-fin Heat Exchangers in Low-temperature Gas Separation Plant, Applications and Design" *British Chemical Engineering*, this issue, p. 18.

[2] Kays, W. M., and London, A. L. "Compact Heat Exchangers", The National Press, California, U.S.A., 1955.

[3] Haselden, G. G. "An Approach to Minimum Power Consumption in Low-temperature Gas Separation", *Trans. Inst. Chem. Engrs.*, 1958, **36**, No. 3, 124.

BCE 6705 for further information

Compressed Air at work in Vauxhall's Luton extension



SEVEN ATLAS COPCO AR9 COMPRESSORS are in use at Vauxhall Motors' Luton factory—recently enlarged and modernised as part of a £36,000,000 expansion plan. These compressors supply air to the body fabrication shops, press shops and other departments, providing power for clutch movements; resetting presses; mechanical handling; loading; welding; and mixing and spraying paint. In addition air is supplied for a number of pneumatic tools such as wrenches, grinders, drills and hoists.

ECONOMIC INSTALLATION

The AR9 compressors were installed at a cost below that estimated for other compressors of the same capacity. The reason being that the AR9 occupies 25% less floor space than is normally required —with consequent economies in compressor house costs.

HIGH OUTPUT

The Atlas Copco AR9 combines thorough reliability of performance with unusually high output per horsepower consumed. The installation at Vauxhall's has a total output of 22,540 c.f.m.

A COMPLETE RANGE OF COMPRESSED AIR EQUIPMENT

Atlas Copco manufactures portable and stationary compressors, rock-drilling equipment, loaders, pneumatic tools and paint-spraying equipment. Sold and serviced by companies or agents in ninety countries throughout the world.

Atlas Copco PUTS COMPRESSED AIR TO WORK FOR THE WORLD

Contact your local company or agent or write to Atlas Copco AB, Stockholm 1, Sweden or Atlas Copco (Great Britain) Limited, Maylands Avenue, Hemel Hempstead, Herts.

C.16

AMERICAN LETTER

Activities in the U.S. Fertiliser Industry

MOST industries have now quite recovered from the U.S. recession of 1957/58, which, in many cases, not only promoted substantial economies in direct and indirect costs (especially the latter) but also appears to have accelerated the development of new products and techniques. The fertiliser industry is particularly active, after the trepidations of the last year or so, both as regard output volume and new technology. A typical example is the rapid establishment of liquid mixed fertilisers. The number of plants in this category is reported to have doubled during the last two years, and well over 300 are now said to be in operation. Although this branch of the industry is believed to have originated on the West Coast, the majority of plants are at present to be found in the North Central States. Current annual output is believed to be in the 500,000 to 600,000 tons range, which amounts to some 4 to 5% of the total output and is expected to rise rapidly.

Liquid mixed fertilisers offer several advantages to the producer, namely, simplicity of manufacture, flexibility and a relatively low plant cost, particularly if distribution and application are undertaken by others. Several advantages are also offered to the farmer, such as low labour costs, bulk delivery and ease of application. However, to compete effectively with high-analysis, solid fertilisers, concentration, solubility and stability must be sufficiently high to justify the use of a mixed solution. Also, the prevention of corrosion in the manufacturing and distribution equipment must be given careful study.

In the majority of cases, liquid mixed fertilisers are based upon reacting furnace-grade phosphoric acid with ammonia under controlled conditions. In another type of operation, neutral solutions of ammonium salts of phosphoric and other acids are cold-mixed in the required proportions. In order to boost the nitrogen content, urea and ammonium nitrate are frequently added. Potassium is usually incorporated in the form of potash, but in certain formulations, such as for tobacco application, potassium hydroxide or carbonate is used. These latter salts also improve overall solubility of the mixed solution.

The problem of concentration is intimately associated with solubility and several stratagems are used, or have been proposed, to achieve maximum results. For instance, a mole ratio of 1.5 to 1.6 for NH_3 to H_3PO_4 may prove helpful in some formulations, in general accordance with the pure NH_3-H_3PO_4-H_2O system. Another method is to add salts in sufficient quantities to form thin slurries or suspensions and stabilise these by chemical and/or mechanical means. The Tennessee Valley Authority investigators report promising results are obtained by using special clays.

Superphosphoric Acid

One important development which may materially affect not only the fertiliser industry, but detergent manufacture and other chemical processes, is the growing availability of superphosphoric acid containing 76% P_2O_5, compared to a customary maximum figure of about 54% for wet-process acid and somewhat higher figures for furnace acid. Use of a 76% P_2O_5 acid gives a large increase in solubility of liquid fertiliser components and so increases both stability and grade. Also recently available is an ammoniated acid containing 11% nitrogen and 33% P_2O_5 by weight for use in cold-mix plants.

Of even greater interest is the effect of superphosphoric acid, or the ammoniated product, on wet-process acid. The latter is notorious for slowly depositing solids and scale during storage and, in addition, neutralisation yields a gelatinous sludge of iron and aluminium salts which are present as impurities in the acid in amounts up to 15%. However, the presence of superphosphoric acid or the ammoniated product has a sequestering effect on the impurities in the wet-process acid. This is believed to be due to the action of pyro and polyphosphoric acids in the superphosphoric acid which suppress precipitation and allow stable solutions to be produced. Typical amounts of superphosphoric acid required have been found to be between 20 and 40% of the total P_2O_5 in the formulation. It is fortuitous that superphosphoric acid is not only one of the least costly sequestrants found to be effective, but at the same time is a large contributor of water-soluble P_2O_5 in the fertiliser solution. Of course, the next step is to try to make a sequestered superphosphoric acid entirely from wet-process acid, and this is presently posing a severe challenge to evaporator makers because of the severe problems of corrosion and heat transfer involved. Already, some U.S. coke oven operators are ammoniating phosphoric acid instead of producing ammonium sulphate because of the more attractive market, and it is understood this practice may prove popular in Europe, particularly if wet-process acid can be used.

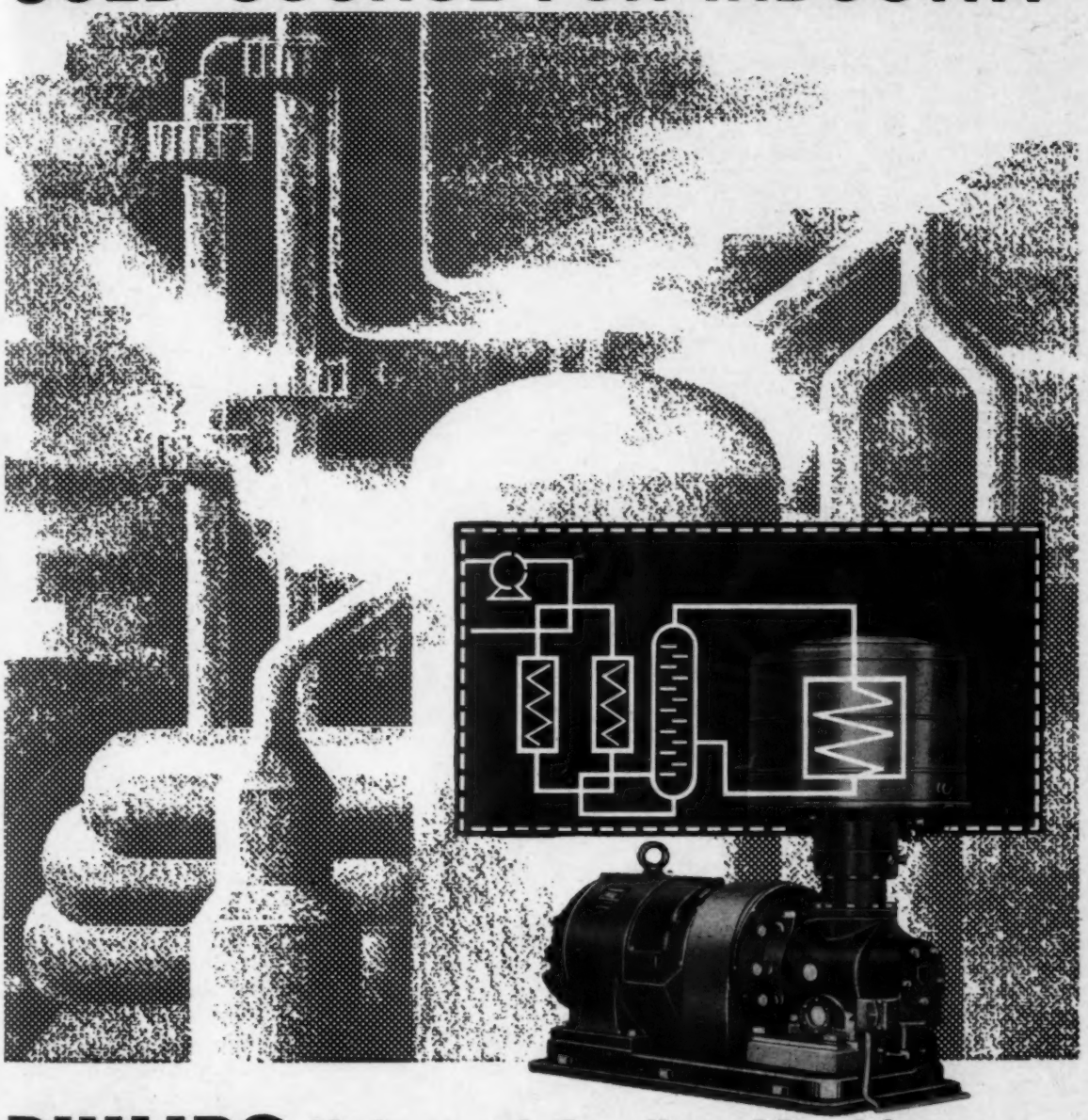
Ammonia is the chief source of nitrogen for both liquid and solid fertilisers, and the development of reliable, partial oxidation processes during recent years for making synthesis gas has enabled both ammonia and urea plants to be built in, or close to, the fertiliser market areas. This is because the raw material, e.g., natural gas, crude oil or refinery residuals can be transported to the site easily and economically, whereas, formerly, many synthesis plants were compelled to locate near a source of solid fuel. The increasing availability of ammonia as a source of nitrogen is reflected in the sharp rise in the U.S. average $N:P_2O_5:K_2O$ ratios, which have changed from 1:2:1.1 in 1950 to 2:2:1.7 in 1958/59. This direction of change is expected to continue, and it now seems apparent that the earlier fears of a huge overcapacity in fixed nitrogen facilities, commenced under earlier government defence plans, were unjustified. Current U.S. nitrogen consumption is now well over an annual figure of 2 million tons.

Present consumption of P_2O_5 is well over 2 million tons a year in the U.S. and most large producers are planning extensive expansion programmes in anticipation of heavy demands from new markets at home and overseas. Phosphoric acid plants with individual capacities of 200,000 tons P_2O_5 per annum by wet processes are being contemplated, and the production of concentrated granular compounds is expected to increase sharply to keep pace with the growing use of liquid mixes. In some large wet-process acid plants, the prevention of atmospheric and stream pollution by fluorine is an increasing problem. Some producers are fortunate in having a ready market for their by-product fluosilicic acid as a source of cryolite for aluminium reduction.

U.S. consumption of potash in fertilisers is now about 1.8 million tons of K_2O annually and some producers believe this figure will not greatly increase during the next few years, because of the anticipated disproportionate increase in nitrogen consumption. However, if a reasonably balanced N:P:K ratio is to be maintained, the U.S. potash figure should also rise appreciably.

BCE 6706 for further information

COLD SOURCE FOR INDUSTRY



PHILIPS Universal Cooling Machine

The Philips Universal Cooling Machine provides low temperature refrigeration wherever you need it in industry. The lowest level that can be reached is of the order of —200°C, as against the normal economic limit of 100°C for any other conventional unit, and the amount of cold generated can be simply varied to suit any particular purpose. This efficient and robust machine has unlimited industrial applications of which just a few are the purification of gases by freezing-out and condensation of contaminants; cold hardening at low temperatures; cold shrinking; and the cooling and condensation of industrially important gases.

OUR ADVISORY SERVICE is always willing to discuss industrial cooling in relation to your own processes. Expert advice is available free of obligation. Write or telephone with your problems or ask for one of our engineers to call on you at your own works.

COMPACT *The machine occupies an overall space of 7 cu. ft. and weighs only 5 cwt.*

VERSATILE *The machine may be easily adapted to suit most applications and special modifications can be designed if required.*

RELIABLE *Superb workmanship coupled with long-term pre-testing has resulted in trouble-free design.*

ECONOMICAL *Low capital cost and proved high operating efficiency.*

Sole distributors in the U.K.: **RESEARCH & CONTROL INSTRUMENTS LTD**

Instrument House · 207 King's Cross Road · London W.C.1 *Telephone: TERminus 2877*

The Philips Universal Cooling Machine is made by N.V. Philips, Eindhoven, Holland for Philips Electrical Ltd. Philips is the trademark of Philips Electrical Ltd.

(RCI0433)

PHYSICAL PROPERTIES OF HYDROGEN AT ATMOSPHERIC PRESSURE

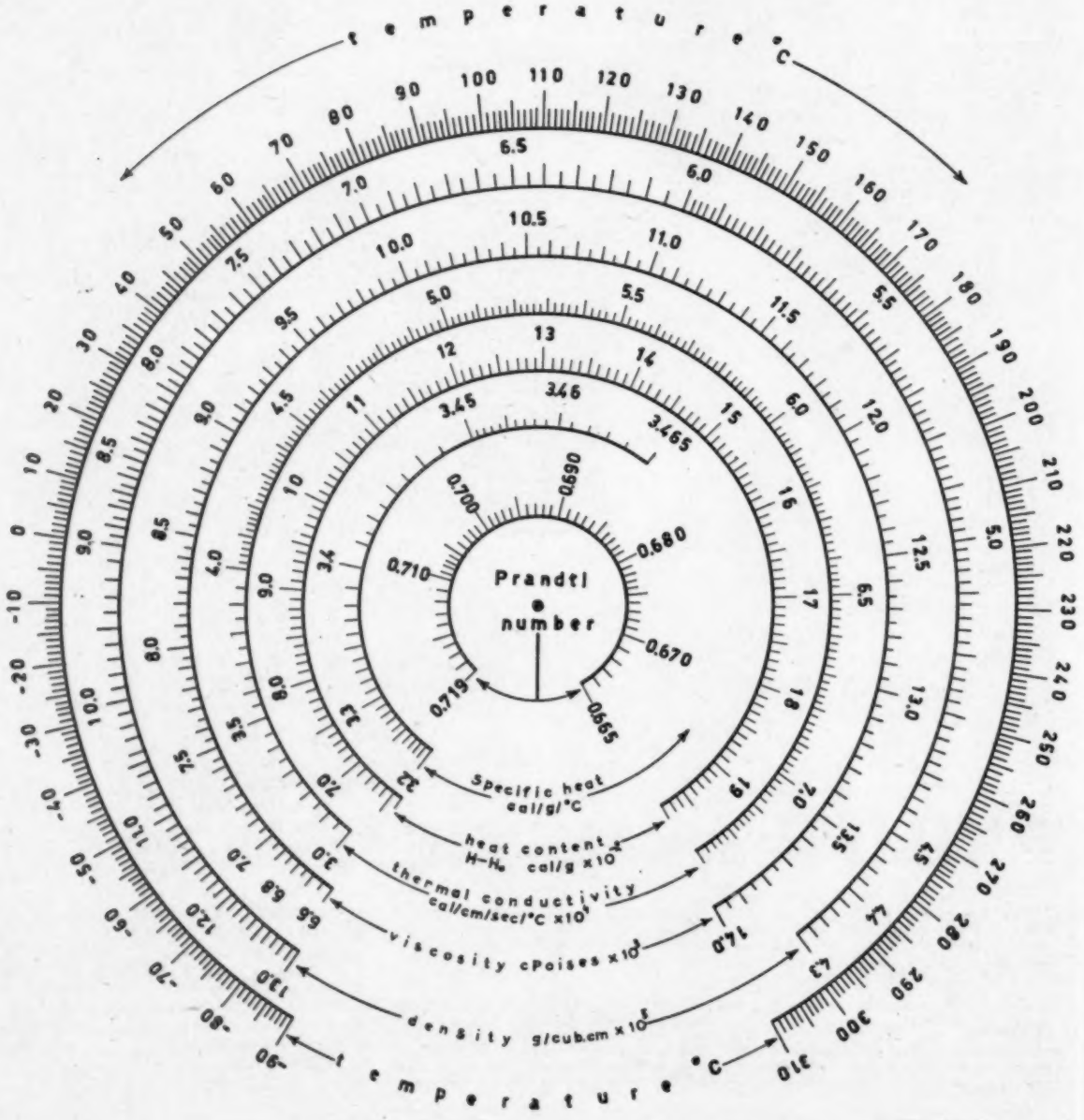
by A. M. P. TANS

The accompanying nomogram has been constructed from literature values. [1] The use of the nomogram is as follows: Any line passing through the centre intersects the scales at corresponding points. The use of circular scales has the advantage that the nomogram is always read off perpendicular to the scales, and, secondly, that a longer scale can be used in a given space.

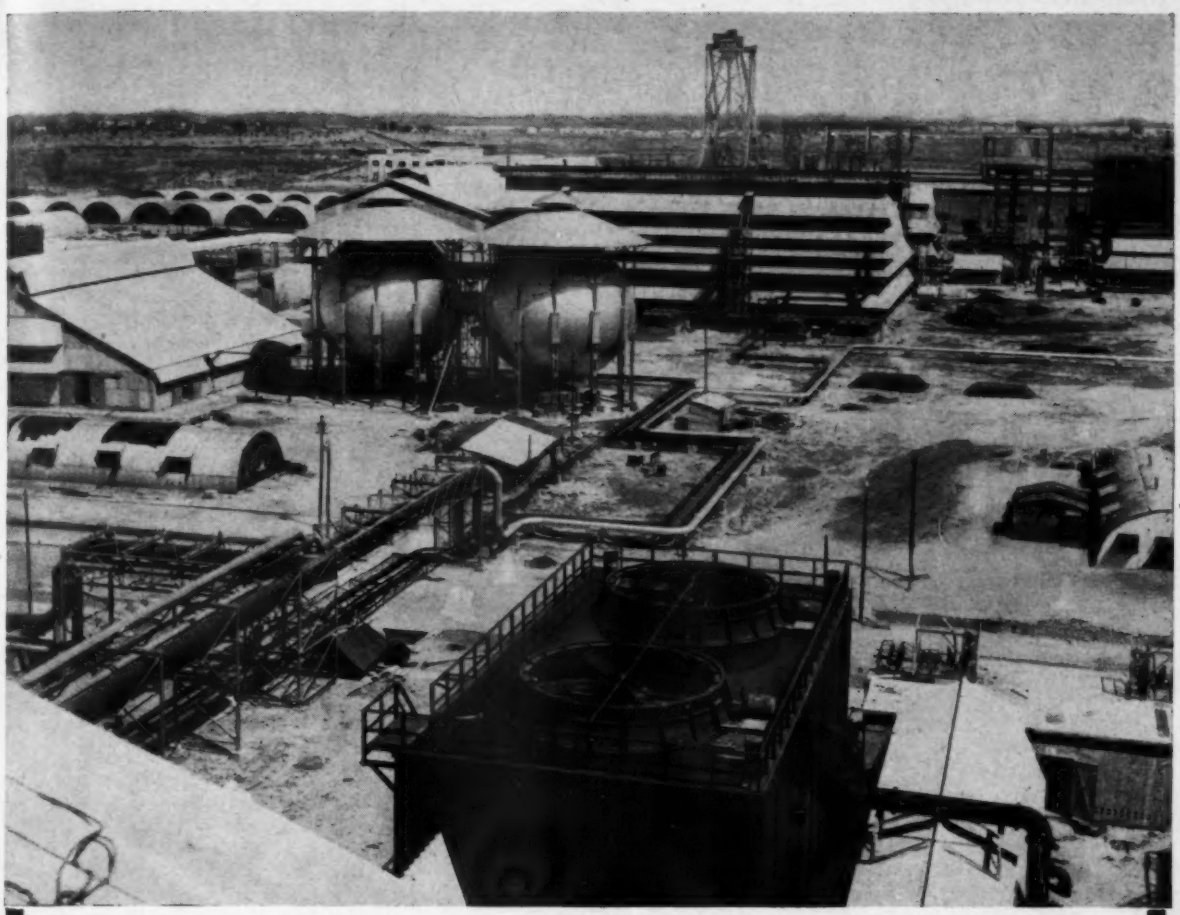
* Central Laboratory, Staatsmijnen in Limburg, Geleen, Netherlands.

REFERENCE

1. Hilsenrath, J. E A. "Tables of Thermal Properties of Gases", National Bureau of Standards, Circular 564, 1955.



British Chemical Engineering
NOMOGRAM NO. 43



$2,000,000,000

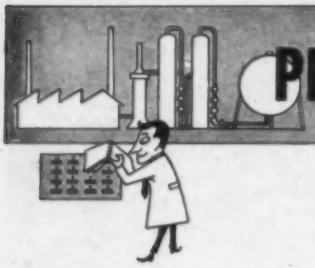
of confidence in CHEMICO

In the years since its founding in 1914 Chemico has completed more than 1200 projects for the Chemical and Petrochemical Industries all over the World. It is estimated that the total value of these contracts represent an aggregate erected value of considerably more than $2,000,000,000. This is an indication of the confidence customers place in the skill and know how of Chemico—one of the largest chemical engineering organizations in the world.

CHEMICO

CHEMICAL CONSTRUCTION (G.B.) LTD.
9 Henrietta Place, London, W.1 Tel: LANGHAM 6571

London headquarters of Chemico.



PROCESS ENGINEER'S NOTEBOOK

Tips for Corrosion Prevention

A COMMON ERROR in condenser design which may lead to severe impingement corrosion is to introduce the hot gases at the top of a vertically mounted unit. If the gases are of a corrosive nature, the tube walls may suffer considerably through the impingement of the gases upon them causing accelerated and high-temperature corrosion. By introducing the hot gases at the base of the condenser, it is possible to take advantage of the wetted-wall falling film formed by the condensing liquid higher up the tubes. The hot gases then impinge upon a liquid film and the corrosive constituents extracted from the gases are diluted by the falling film of liquid. This practice will not only reduce corrosion considerably, but it is likely to lead to higher heat-transfer rates because the temperature driving force between liquid film and hot gas will be greater than in the case of the less-favoured arrangement where the hot gases are in contact with a dry metal surface.

Autoclave design is another field where improvements in design can lead to improved resistance to corrosive conditions. In the design of nozzles, if their ends are left square, capillary action may raise the corrosive liquors around the nozzle surfaces into crevices at the openings and into discontinuities in the runs of the weld metal. As a result, concentration cell corrosion often takes place. Again, the square-end nozzle may cause drainage down the side wall, thereby producing a groove of corrosion at that part of the vessel. By providing a bevelled surface at the lower end of the nozzle a more positive drainage is obtained, and rise of the liquids through capillary action is hindered.

Protecting Steel Sulphonators

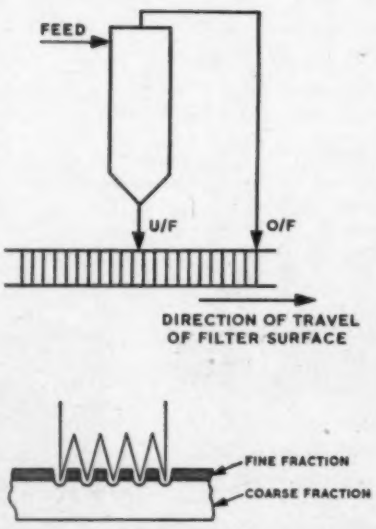
Steel sulphonators are also known to suffer corrosion at the liquid-vapour interface. This can happen where sulphuric acid is used for sulphonation if the acid is added to the surface of the organic liquid in the vessel while agitation is taking place. The dilution effect and the temperature rise brought about by the addition causes the walls of the vessel to come into contact with hot dilute sulphuric acid, with the result that severe corrosion takes place. In one case this type of attack was reduced by the simple expedient of extending the sulphuric acid inlet pipe below the surface of the sulphonation mass and at a point in the vessel where the intensity of agitation was at its greatest. The vessel wall is then not in direct contact with dilute sulphuric acid. In another case this method was not successful because foaming occurred. The solution adopted consisted of introducing the mixed mass of reactants into the autoclave after they had passed through a jet-mixer constructed of a material capable of resisting the action of the hot dilute sulphuric acid.

How to Reduce Filter Cake Moisture

QUITE OFTEN a lower filter cake moisture is desired than can be obtained by conventional means—increasing the drying portion of the filter cycle, operating with a thinner cake, increasing the vacuum pump capacity.

In metallurgical bottom-feed vacuum filtration a waste-product siliceous coarse-grained material may be applied as a pre-coat, at the same time eliminating the air blow from the filter, which normally assists cake discharge. Sand is usually removed early in the flowsheet and, being a waste product, can be used for "back-filling" underground.

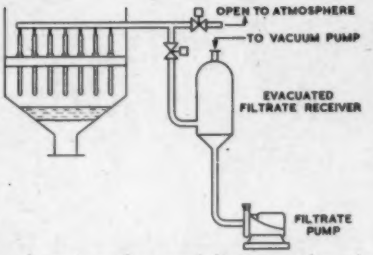
Filtration of a coarse-grained crystal would normally be carried out by a top-feed drum or horizontal pan-type filter. The same principle may be applied as above, i.e., by splitting the crystals into two fractions using a liquid cyclone (see BRITISH CHEMICAL ENGINEERING, Vol. 1, No. 1, article by R. H. Naylor), in which a coarse pre-coat matrix is formed.



If the fine fraction contains a considerable amount of fines, which forms an effective seal on top of the filter cake when filtering in a horizontal plane, thereby preventing the passage of air under the vacuum differential, then this "surface" should be broken up with a stationary rake. The penetration of the raking fingers should only extend to a depth just below that of the fines, leaving undisturbed the coarse fraction. Movement of the filter matrix through the rake produces furrows which permits the air to displace the entrained water in the filter bed. The lower diagram (left) shows the principle of operation of the stationary rake and how it extends to a depth just below that of the fines.

Suspension Separation

THE PROCESS of continuous filtration of suspensions and slurries can be speeded up by including a preliminary thickening operation in the filtration scheme in series with a centrifuge or a rotary vacuum filter. For the purpose a box-type filter can be used of the type shown in the diagram. It consists of a trough which terminates at its lower end in a conical or pyramidal base for collecting the thickened precipitate. The trough contains an assembly of filter elements or plates which are connected to a common conduit for the passage of the filtrate. This conduit has two branches, one being connected through an intermediate filtrate receiver to the filtrate pump. The other branch can, if required, be open to atmosphere. In each branch a magnetic valve is fitted; when filtration is in progress the valve in the connection to the receiver is kept open and the valve in the side branch is closed. Filtration is maintained until the cake resistance becomes excessive. The valve in the side branch can then be made to open after the receiver valve has been closed. A time switch can be used for obtaining the proper sequence of valve operation. Through the opening of the side branch valve,



air enters the conduit connecting the filter plates and forces the filtrate remaining in the conduit back into the trough of the filter. At the same time, through the lifting force exerted on the cake, the latter drops into the base of the trough in a compact form. While still in this condition the precipitate can be removed by means of a diaphragm or screw pump and fed to a continuous rotary vacuum filter or a centrifuge. The timer can then close the side branch valve and open the receiver valve to allow the filtration in the box filter to proceed. Such a combination of box filter in series with another type of filter may often prove to be an economical solution to the separation of difficult-to-filter suspensions. The above arrangement was described in an article by H. Rumpelt, *Chem. Ing. Tech.*, 1958, 30, 226.

Lagging Mattresses

THE ORTHODOX METHOD of mattress making is to cut and sew up an envelope of the required size and shape, leaving a hole for filling at a suitable point. Insulating material is then packed into the envelope and the filling hole stitched up; in the case of certain irregular shapes, this is still the best method.

Improved Lagging Method

The improved method is as follows:

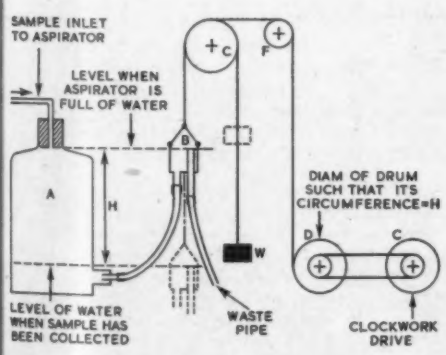
(1) the wire netting is cut to the required size and the edges turned up to form a shallow box, approximately 1½ in. deep;
(2) the slag wool is placed in this box and built up to the required thickness and density;
(3) a second piece of wire netting is placed on top of the slag wool and the edges twisted in to those of the box. The slag wool is now totally enclosed in a wire netting box; and
(4) asbestos or similar cloth is now placed over one side of the mattress only and stitched to the netting.

Apart from the saving in material by this method, the manufacturing time is reduced and also a much more uniform mattress is made.

(Courtesy of I.C.I. Wilton Services Works)

Constant-rate Variable-period Gas Sampler

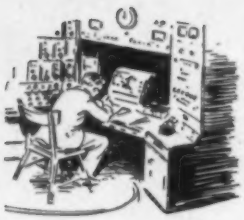
BY USING THIS APPARATUS it is possible to collect samples of gas over any specified period of time at a constant rate. This is particularly valuable when it is required that the sample is to be a true average of the whole. The device consists of an aspirator, A, with a tightly fitting bung and a sample inlet pipe. A water overflow weir, B, is connected to the aspirator by a rubber tube. By lowering B, the level of water in the aspirator falls and the gas sample enters. The water overflow weir is lowered at a constant rate by means of the driving mechanism which consists of a drum, D, the circumference of which is equal to H, the distance the overflow weir falls when collecting the sample. The weir is connected to the drum by a light string over pulleys C and F. A clockwork mechanism (making one revolution in 24 hr—a discarded pressure recorder would be ideal) drives the drum. The rate of sampling may be varied by fitting the drum and driving mechanism with pulleys of various diameters. The inertia of the weir is counter-balanced by fixing a counter weight on pulley C. Should the sample be soluble in water, then a protective film of light oil, such as liquid paraffin, can be placed on the water in the aspirator.



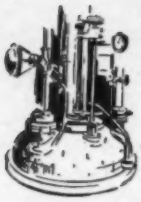
R.H.U.

Contributions invited

THE EDITOR invites readers to submit, for possible publication in "Process Engineer's Notebook", notes on practical devices that have been elaborated on the job for improving the operation or increasing the safety of processes, or that have been extemporised in special circumstances. The contributions preferably should be of about 250-500 words, and be illustrated with a line diagram or sketch, or a photograph. A minimum of three guineas will be paid for each contribution used. They will be published anonymously or under the contributor's name, according to his wishes.



LETTERS TO THE EDITOR



Tubular Versus Polybloc Exchangers for Pickle Liquor

Dear Sir:

We have noted Mr. G. H. Black's comments in the December, 1959, issue of BRITISH CHEMICAL ENGINEERING on our article "A Novel Application for Graphite Heat Exchangers" published in the July, 1959, issue of BRITISH CHEMICAL ENGINEERING. We are, of course, aware of the merits of Keebush material for the construction of pumps and pipework. However, we do not share his opinion on the merits of tubular heaters for pickling liquor.

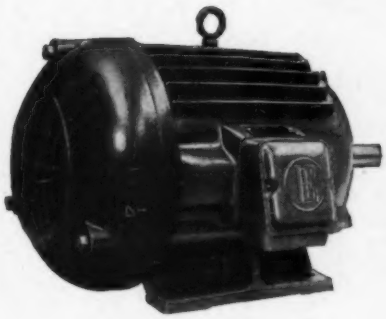
The tubular system was developed over the years for constructions in metals for which it is suitable, because most metals are isotropic. Conversely, graphite, which is the most important non-metallic constructional material for corrosion-resistant heat exchangers, is highly anisotropic, and whereas its compressive strength is very good, its tensile and flexural strength are poor. Structures based on tubular or such-like systems fail to exploit the good compressive strength of graphite, for they are subject to tensile and flexural stresses. It appears, therefore, not advantageous to use such material in the form of long thin shapes such as tubes. Any preferred orientation of the anisotropic graphite crystals is, in tubes, opposite to the direction of the desired heat transfer. To exploit the thermal conductivity of graphite to best advantage, it is required to use drilled monolithic blocks. To fix the tubes in the header plates, cemented joints are generally used which have to be not only chemically inert, but which are also subject to differential expansion stresses. Normally, the limiting working pressure with tubular units is only a fraction of that permissible with graphite blocks. These and other characteristics of the material raise the question whether there exists a constructional method which is less suitable for graphite than is the tubular system? In a paper entitled "Considerations on the Design of Graphite Heat Exchangers" which was presented before the Chemical Engineering Conference at Oslo in October, 1958, and reprinted in BRITISH CHEMICAL ENGINEERING of March, 1959, one of the authors explained his reasons for the development of a constructional system (known as the Polybloc system), which we consider more suitable for graphite constructions, because it minimises the effect of the less favourable properties of the material, and takes full advantage of its many favourable characteristics. It is also a major merit of the Polybloc system that the short length/diameter passage ratio which it involves takes advantage of the end effect to inhibit streamline flow at comparatively slow flow rates. This induces not only improved heat-exchange efficiency, but introduces an important scale-inhibiting effect. It is recalled that, at the Wednesbury plant described in our aforesaid article, the Polyblocs continue to perform at the originally measured heat-exchange coefficient, and that the first Polybloc there has been in substantially continuous operation for 16 hr per day since August, 1957. Neither this Polybloc, nor the two others subsequently installed (January, 1958), ever required any maintenance or other attention whatsoever. This illustrates the very real and practical importance of the anti-fouling effect of our system. Conversely, we understand that tubular units, if used for the heating of pickling liquor, may suffer loss of heat-exchange efficiency through fouling up, after comparatively short periods of operation, and may require frequent cleaning out. The mechanical descaling of long graphite tubes (Mr. Black mentions 9 ft length) seems, in our opinion, a rather delicate operation, and it appears difficult to visualise how this can be done without damage to the tubes. We feel that it is preferable to employ a system which can avoid the trouble and expense of frequent cleaning and maintenance.

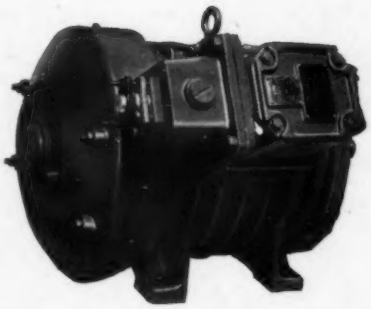
It is recalled that in the plant described in our article this is achieved with a pickling liquor flowrate through the heat exchangers of only 2 ft per sec., and a pressure drop which is so small that it is hardly measurable. Mr. Black does not cite flow speed figures. In our opinion, an excessive liquor flow speed would be necessary to obtain a similar anti-fouling effect with heat-exchanger tubes of 9 ft length and normal diameter, involving heavy wear and maintenance cost of the pumps. Conversely, to approach the L/D ratio of the process passages of the aforesaid Polyblocs, the i.d. of such tubes would have to be about 10 in., which appears economically impracticable. It would be interesting to learn at what pickling liquor flow speed Mr. Black can guarantee complete and permanent inhibition of iron sulphate scale deposits, with the 9-ft-long heat-exchange passages which he cited? Figures showing overall heat-exchange coefficients as a function of pickling liquor flow speed and pressure drop and variation of these coefficients (if any) from year to year (provided of course, that like the Polyblocs at Wednesbury, the tube bundles never require any cleaning) would also be of considerable interest for comparison with the figures published in our aforesaid article. The heat exchangers made from 9-ft-length graphite tubes which Mr. Black cites occupy, presumably a considerable space. Conversely, it is recalled that the extreme dimensions of the Polyblocs at the Wednesbury plant are only 9½ in. in diameter and 2 ft 8 in. high.

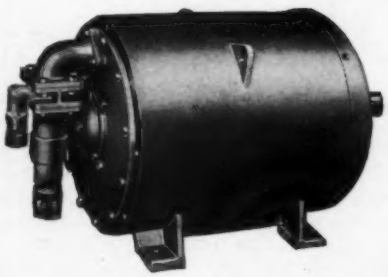
A major object in replacing live steam injection and/or submerged coils by external heat exchangers is the savings in fuel, maintenance and operating costs. The provision of corresponding figures from steelworks using graphite tubular heaters would greatly assist in clarifying the questions which were raised, as they would permit some measure of quantitative comparison with the important savings which we were able to record. As our article cites figures actually reported by an engineering works using Polyblocs for the pickling of steel, it appears reasonable that they should be compared with corresponding figures from a works using tubular graphite units for a similar duty.

Concerning the last paragraph of Mr. Black's letter, it is noted that this refers to a graphite tube bundle for the heating of pickling liquor in a continuous strip pickling line. He does not cite the quantity of pickling liquor in the tank, quantity of steel strip pickled per hour, and whether the tank is covered? Details of these and other data are essential for any comparison. Whereas in our experience the pickling liquor temperatures for intermittent pickling (ingots, coils, etc.) are usually around 60-75°C, much higher pickling liquor temperatures are normally imposed for continuous strip lines. To increase pickling speed as a function of strip throughout, and to reduce pickling cost as well as obtain increased production, makers prefer generally to work at strip pickling liquor temperatures as near as possible to boiling point. This would not be advantageous with steam of less than 25 psig as mentioned by Mr. Black, because the steam temperatures would then be less than 115°C, resulting in somewhat poor $\triangle t_m$. In our experience, managers of continuous-strip pickling plant prefer to work (for speed and efficiency) at pickling liquor temperatures up to 100°C and heating steam pressures up to 85 psig (steam temperatures up to 158°C for favourable $\triangle t_m$). Such steam pressures are fully compatible with graphite units based on the Polybloc system. It appears

Motors for corrosive and explosive atmospheres







CHEMICALS & PAINT

Class 'LYC' motors to B.S. 2083 up to 25 h.p.
'ENGLISH ELECTRIC' is the only Company making chemical works motors to this standard. Tens of thousands of British Standard motors are in service. Available up to 50 h.p. in N.E.M.A. sizes.

Write for Publication DM/206

PETROLEUM

Class 'XLK' flameproof motors Squirrel-cage or Slipring. Totally-enclosed fan-cooled or Totally-enclosed. Complying with Oil Companies Materials Committee requirements. Buxton certified.

Write for Publication DM/154

COAL MINING

Class 'LP' steel-barrel, flameproof Squirrel-cage motors. Totally-enclosed fan-cooled available up to 150 h.p. Totally-enclosed available up to 50 h.p. Buxton certified for underground use.

Write for Publication DM/225

AVAILABLE FROM STOCK

'ENGLISH ELECTRIC'

industrial motors

THE ENGLISH ELECTRIC COMPANY LIMITED, MARCONI HOUSE, STRAND, LONDON, W.C.2.
Industrial Motor Works, Bradford
WORKS: STAFFORD · PRESTON · RUGBY · BRADFORD · LIVERPOOL · ACCRINGTON
DM. 74

difficult to visualise how they could be employed with graphite tubular units which, we understand, are normally specified for working pressures which are only a fraction of our aforesaid figures.

Some degree of rough handling of equipment is often unavoidable in heavy industries such as steelworks. This is a reason for the compact design of the Polybloc elements. Experience has shown that the effective robustness of Polyblocs is comparable to that of units constructed completely from metal. However, it appears difficult to visualise how long, thin graphite tubes would stand up to transmitted shock, e.g., caused by the accidental impact on the shell of an ingot or such-like charge.

We feel that as some of the views expressed in Mr. Black's letter differ from our own experience, it would be of service to us and others if they could be elucidated by actual quantitative performance data, measuring in works using tubular graphite heat exchanges for steel pickling liquor, both in intermittent and continuous processes.

Yours faithfully,

A. HILLIARD
N. H. RICHES
B. A. GARMAN

Société Le Carbone-Lorraine,
64 Finsbury Pavement,
London, E.C.2.

Gas Purification Plant Oxidiser and Purifier

Dear Sir:

The note appearing on page 481 of the August-September, 1959, issue of BRITISH CHEMICAL ENGINEERING under the above title describes a defrothing method and equipment used in connection with a gas-purification process known as the Manchester Process. This is the subject of a number of patents, in particular B.P. Nos. 660,671 and 667,963. The plant referred to was in fact designed and constructed for the Wales Gas Board by Humphreys & Glasgow Ltd.

The Manchester Purification Process is a process for removing hydrogen sulphide from gases by a particular method of washing with a suspension of a suitable iron compound in an alkaline carbonate solution. We and certain other companies have licences to design and build plant for the process. A number of plants have been built, differing in various respects. For example, the plants built by the other licensees use arrangements for destroying froth which are completely different from the defrother which we designed for the plant which we built at Cardiff. Therefore, although our defrother is very effective and very useful, it is not essential to the Manchester Process. Also, of course, the defrother is capable of use in other processes where defrothing is required.

We are anxious, in view of the absence of a reference to the patent position in your description of the defrothing method detailed in your Process Engineer's Notebook, that your readers should not have the impression that they are free to use this method of defrothing either for the Manchester Process or any other application.

Yours faithfully,

A. J. WOODIN

Humphreys & Glasgow Ltd.,
22 Carlisle Place,
London, S.W.1.

French Industries Cut Relining Costs

LININGS of neoprene are being used effectively in many French industries to protect equipment and vessels from a wide range of corrosive influences. The result is a drastic reduction in costly relining operations. Neoprene was introduced a little more than 27 years ago in the United States as the first commercially available multi-purpose synthetic rubber. It has a number of outstanding properties which make it highly desirable as a lining or coating material. Because of these properties, neoprene-lined vessels can handle products which would quickly attack natural rubber or the less durable lining materials. Neoprene offers resistance to chemicals, oils, abrasion, temperature changes, ozone and weathering. The following case histories*, from several different French industries, illustrate how these properties of neoprene are being utilised to advantage in lining industrial equipment:

Neoprene linings are being used in vats for washing abrasives because of their resistance to heat, chemicals and abrasion. A sulphuric acid solution bath is used, heated to 80°C by the injection of live steam. The abrasive particles are agitated by the steam pressure, with four separate batches of abrasives being washed each 24 hr.

The 4-mm-thick neoprene lining used in these abrasive washing tanks replaces a 15-mm coating of lead used previously. The lead coating eroded away and had to be replaced after only six months to a year's use. The neoprene lining was applied three years ago and is still in good condition.

Neoprene's resistance to rapid temperature changes is used to good advantage in pigments manufacturing. It is the only lining material used to date which can withstand the sudden temperature changes from boiling to freezing necessary in this operation. The products in this vat also are subject to agitation.

The coatings in vats for processing uranium, titanium, vanadium and cobalt must simultaneously resist attack by chemicals and abrasion. The ores are treated with sulphuric acid at a temperature of 70°C. A 5-10% solution of nitric acid is added to complete the processing. The giant agitators used in this process as well as the vessel walls are coated with neoprene 6 mm thick.

Large vats of 200 cu. m. are lined with neoprene sheet over a thin base of polyisobutylene. The neoprene, unlike the natural rubber and other materials tried previously, resists traces of the organic solvent present in this vat. The polyisobutylene prevents the hydrochloric acid from being diffused through the neoprene to the surface of the tank. The lining in this vat has been in use since 1957 and is still in good order.

The conical floors, interior sidewalls, armatures and sectional iron supports of a three-storey-high Dorr Oliver thickener (a device used to take water from chemical slurries) are lined with more than 3000 sq. m. of sheet neoprene. Use of neoprene for lining this mammoth piece of equipment represents one of the largest single applications for a neoprene lining in France.

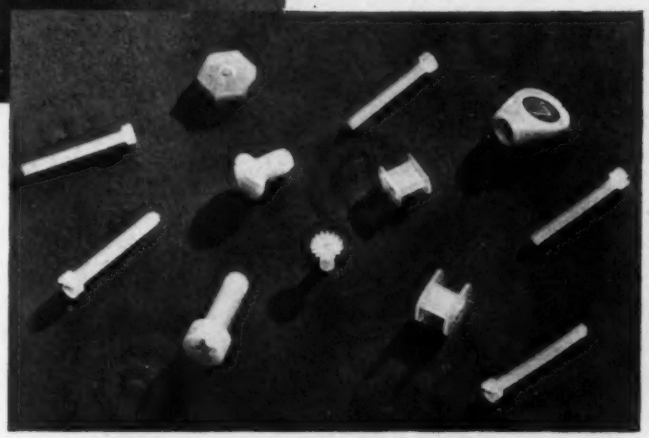
The neoprene linings used for these applications ranged in thickness from 2-6 mm, depending upon each specific area being lined. They were applied to the interior of various vessels at the plant site with an adhesive, after the steel or concrete surfaces had been sanded smooth. For smaller vessels unvulcanised sheets are bonded to the metal interiors, then cured by autoclaving. Where the vessel is too large for autoclaving, the sheet neoprene is vulcanised before application in the plant of the lining manufacturer.

The lining used had the following properties: density, 1.38; shore hardness A, 60; elongation at break, 500-550; breaking strength, 150-160 kg/sq. cm; and modulus at 200%, 45-50 kg/sq. cm.

*The case histories refer to linings applied by Société Chimique de Gerland, 49 rue de la République 49, Lyon, France.

Today's engineers are using the material of tomorrow – 'Maranyl' A 100 nylon

Selection of screws and components moulded in 'Maranyl' A 100 nylon by George Goodman Ltd., Robin Hood Lane, Birmingham, 28



Screws and components made from 'Maranyl' A 100 nylon are light in weight, corrosion resistant and good electrical insulators

The screws and components illustrated were moulded in 'Maranyl' A 100 nylon by George Goodman Ltd., Robin Hood Lane, Birmingham, 28. They are light yet strong, resistant to corrosion and are good electrical insulators. They are long-lasting and resist vibrational fatigue and abrasion.

There are many other important features of this I.C.I. plastic that commend it for a multitude of uses in the industry of today — and tomorrow. It resists oil, fuel, greases and most common solvents. It needs no lubrication. Its inertia is low. It is easy to fabricate. It has low coefficients of friction and is silent in operation. 'Maranyl' A 100 nylon is available in a wide range of colours.

I.C.I. RESEARCH AND TECHNICAL SERVICE

'Maranyl' A 100 nylon is backed by the full force of I.C.I. Research and Technical Service. That service is available to you. If there is anything you wish to know about nylon and its uses; if you have a problem concerning noise, lubrication, wear and durabilty — consult I.C.I. They will be glad to help.

'Maranyl' A 100 NYLON

'Maranyl' is the registered trade mark for the nylon compound manufactured by I.C.I.

ICI

IMPERIAL CHEMICAL INDUSTRIES LIMITED · LONDON · SW1

PN 155

Book Reviews

Progress in Cryogenics—Volume 1
Edited by K. Mendelssohn
Heywood & Co. Ltd., London, 1959, 63s.

THE techniques employed in low-temperature physics, together with the practical applications of some of the phenomena which this subject has revealed, are now loosely classified by the word "cryogenics". The scope is wide, as is evident from the topics covered in this first progress volume. Due to the eminence of the editor and his wide range of interests and international contacts, a very strong team of contributors has been brought together.

D. R. YOUNG, of the I.B.M. Research Laboratory, New York, leads with a discussion of computer circuits using superconducting elements. He describes the various types of switching and sensing devices and the ways in which these are linked in modulator and memory circuits.

Thermo-electric cooling, which is a borderline subject in cryogenics, is reviewed by D. A. WRIGHT, of the General Electric Co., Wembley. His article covers some of the basic physics of semiconductors and the problems involved in selecting suitable pairs for thermo-electric refrigerators.

Two technical aspects of low temperatures are covered respectively by M. M. FULKS, who writes with the authority of the National Bureau of Standards on evacuated powder lagging, and B. R. BROWN, of the British Oxygen Co., who discusses plate-efficiency in the fractionation of liquefied gases.

Four papers deal with physical measurements at low temperatures. H. M. ROSENBURG, Clarendon Laboratory, reviews apparatus for measuring the mechanical properties of metals, whilst ultrasonic attenuation in metals is described by R. W. MORSE (Brown University, U.S.A.). A Russian contributor, N. V. ZAVARITSKY, deals with the determination of specific heats by the temperature-wave method. General methods of low-temperature calorimetry are summarised by R. W. HILL (Clarendon Laboratory).

The remaining topic is that of frozen free radicals. G. J. MINKOFF contributes an extensive and well-documented review of the present state of the subject—largely arising from his association with research at the Bureau of Standards in Washington.

The standard of writing and presentation is consistently high and each paper has an ample bibliography.

G.G.H.

DECHEMA Monographien.
Band 32, Nr. 451-476. Band 33, Nr. 477-502. Deutsche Gesellschaft für Chemisches Apparatewesen E. V., Frankfurt A. M., 1959

THOSE who attended the ACHEMA exhibition in Frankfurt in 1958 will remember that several series of lectures and discussions were held at the same time as the DECHEMA exhibition of chemical plant and machinery. Conscientious attendance at these, as well as at the exhibition, the works visits and the many social functions, contrived to make ACHEMA 1958 a very exhausting week for the participants. Those lectures given under the auspices of the European Congress of Chemical Engineering are now published as DECHEMA monographs, Band 32 and Band 33. The three languages of the Congress were German, English and French, and when a lecture was given in one language there was simultaneous presentation of the text in the other two languages. In the published versions, the lectures are given in full in German, with summaries in English and French. There is no report of any of the discussions which followed the presentation of the various papers.

The publishers have divided the subject-matter into 26 papers concerned with foundations and principles, ("Betriebstechnik—Grundlagen") in Band 32, and 26 papers describing processes and chemical plant ("Betriebstechnik—Verfahren und Operationen") in Band 33. As examples of the kind of distribution, such papers as VAN KREVELEN'S on "Advances in the Control of Technical Reactions", papers on physical properties, on heat and mass transfer and on mixing, etc., are in Band 32. Included also is a paper by S. KAMEI, of Kyoto University, which the publishers seem particularly pleased to have; they mention in their preface that this gives important experimental results on the drying of many materials which previously had been published only in Japanese language journals. Band 33 begins with one of the "prestige lectures", "Acetic Acid Recovery", by D. F. OTHMER, and includes papers on manufacturing processes, on production operations (for example, the electrolysis of water under pressure) and on the design and performance of equipment (for example, air-cooled heat exchangers). The two volumes comprise a useful collection of largely review papers which are particularly interesting because of their international origin. It is perhaps unfortunate that the whole of the text is not published in English and French versions as well as German.

The lectures were given at a time when German industry was trying to comprehend just what unique function was possessed by chemical engineers; it was still seen as a "coming to terms" of the functions of the chemist and the engineer. Some discussion of this is given in the preface to Band 32. On reading this, one has the feeling that when the Germans have finished sorting this out they may be left with an interpretation which is more flexible than that of the Institution of Chemical Engineers. The Germans certainly seem to recognise that the diversification of talent which is met in modern chemical industry is not likely to be fostered by any stereotyped qualification requirement.

C. S. SILBUS

The Structure and Properties of Porous Materials, being the proceedings of the tenth symposium of the Colston Research Society held in the University of Bristol, March, 1958.
Edited by D. H. Everett and F. S. Stone
Butterworths Scientific Publications, London, and Academic Press Inc., New York, 1958, 389 pp., 60s.

COLSTON RESEARCH SOCIETY cover a large range of subjects: biochemistry and biophysics, sociology and psychology. In 1958 the Society held a symposium on the theoretical and practical aspects of porosity. Outstanding workers from 11 countries, including a number from Russia, attended this conference, the complete proceedings of which are given in this book. The papers presented comprise reviews of fundamental research on the structure of porous crystals, gels and carbons; theoretical treatments of the behaviour to be expected of models of porous systems; accounts of the nature of porous structures in paper and materials of building construction; and also illustrations of the significance of porosity in systems of organic origin.

Titles of the papers were "Aspects of Sorption in Porous Crystals" (R. M. BARRER); "The Surface Characteristics of Carbons" (W. F. K. WYNN-JONES); "Energetics of Sorption" (G. L. KINGTON); "The Shapes of Capillaries" (J. H. DE BOER); "Some Problems in the Investigation of Porosity by Adsorption Methods" (D. H. EVERETT); "The Flow of Low-boiling Gases through Saran Charcoal" (J. R. DACEY and J. A. FENDLEY); "Statistical Descriptions, Correlation Factors and Physical Properties of Granular and Porous Solids" (E. A. FLOOD); "The Relation Between Gas Permeability and Pore Structure of Solids" (P. C. K. WIGGS); "The Preparation and Adsorptive Properties of Silica Gels" (A. G. FOSTER and J. M. THORP); "The Structure of Certain Corpuscular Adsorbents and its Influence on their Adsorptive Properties" (A. V. KISLEV); "The Influence of Surface Charges on the Properties of Porous Materials" (R. K. SCHOFIELD); "The Influence of the Adsorption of Cations on the Texture and Structure of Silica Gels" (C. MOUGEY, J. FRANCOIS-ROSETTI and B. IMELEK); "The Structure of Porous Metal Films and its Influence on Chemisorption Properties (N. N. KAVTARADZE); 'The Porous Structure of Paper" (H. F. RANCE); "The Interaction of Swelling Agents with Wool and Nylon" (G. KING); "The Structure of Porous Building Stone and its Relation to Weathering Behaviour" (D. B. HONEYBORNE and P. B. HARRIS); "The Ascent of Sap and the Movement of Soluble Carbohydrates in the Stems of Higher Plants" (R. D. PRESTON).

The discussion sections were particularly interesting, as most recent data and experimental techniques were presented there, and the comment was very lively. The review papers at the beginning of the volume will probably find the most readers, as they are authoritative statements of our knowledge of the properties of porous structures today.

W. STRAUSS

BCE 6710 for further information

CHAPMAN & HALL

★ *New British & American Books* ★

LABORATORY INSTRUMENTS

Their Design and Application

by

A. ELLIOTT, PH.D., D.SC.

and

J. HOME DICKSON, M.SC.

SECOND EDITION

530 pages *Illustrated* *55s. net*

This book, based on the experience of many years of research work, presents the principles on which good design is based, and gives many examples of practical applications. It was an immediate success when it was first published, and the demand for a second edition has given the authors the opportunity of revising the book completely and of adding much new material.

★

From John Wiley & Sons, Inc.

CHEMICAL PROCESS PRINCIPLES

Part II — Thermodynamics

by

OLAF A. HOUGEN
KENNETH M. WATSON
ROLAND A. RAGATZ

SECOND EDITION

624 pages *Illustrated* *78s. net*

This new edition of the second part of a three volume work brings together the theoretical development and practical applications of the basic engineering science of thermodynamics, plus advanced treatment of the more specialized developments of chemical engineering thermodynamics. Carefully revised to reflect the most recent technological advances, the book contains a great deal of new material.

★

From Reinhold Publishing Corporation

SOURCE BOOK OF THE NEW PLASTICS

(Volume I)

by

HERBERT R. SIMONDS

362 pages *Illustrated* *80s. net*

A comprehensive review of the new plastics: their production, applications and selection. Sixty plastics producing companies in the U.S.A. contribute articles describing their materials—including the seven new primary plastics of 1958. The book is a unique approach to plastics that concerns itself almost exclusively with latest developments. Whatever the reader's question, need or interest in new plastic materials, it is dealt with in a specific, accurate and practical manner.

37 ESSEX STREET, LONDON, W.C.2

BCE 6711 for further information

PROTECT CHEMICAL PLANT AND MACHINERY WITH . . .

"EPINAMEL"

ANTI CORROSION COATINGS

This illustration, reproduced by permission of Shell Chemical Co. Ltd., presents a typical example of the successful application of Epinamel Anti-Corrosion Coatings to Chemical plant.

Write for full details of Epinamel to :

ARTHUR HOLDEN & SONS LTD.
BORDESLEY GREEN ROAD, BIRMINGHAM, 11
TELEPHONE : VICTORIA 2761 (5 LINES)

World of Chemicals

D.L.F. Aid for Jordan

A loan agreement has been signed in Amman, Jordan, under which the U.S. Government will lend $1,500,000 to the Jordan Phosphate Mines Co. Ltd. to help finance expansion of its operations. The loan will permit expansion of the output of the company's mines near Ruseifa, north-east of Amman, from the present output of 320,000 tons of dry phosphate a year to some 600,000 tons a year in the first phase. Ultimately, if markets and other economic factors warrant it, the output will be increased to about 1,050,000 tons a year. The Ruseifa deposits and reserves are extensive and provide the country's major opportunity for expanding its exports. Thus, it is expected that the operations made possible by the Development Loan Fund will increase employment, foreign exchange earnings and Government revenues in Jordan.

One-step Catalytic Process

A dual-purpose one-step catalytic process for purification and dealkylation of aromatic light oils has been developed by Houdry Process Corp., in co-operation with a large eastern steel company. Available for general licensing to coke oven operators and petroleum and chemical companies, the new LITOL process is reported to be of particular interest to coke oven operators. It is used for the processing of aromatic light oils to obtain benzene, toluene and xylene which meet the most stringent purification specifications. The purified benzene obtained contains less than 1 ppm of thiophene and has a freezing point of 5.4°C, or higher. The Houdry process is said to eliminate the need for a second step involving either costly azeotropic distillation or solvent extraction.

Haifa Fertiliser Research

In their annual report for the year ended March, 1959, the directors of Fertilisers & Chemicals Ltd., Haifa, stated that the company's research division have been engaged in the development of new liquid and compound fertilisers and a broad range of industrial chemicals for sale at home and abroad. Several new phosphate salts based on local raw materials were developed and commercial production of these had been commenced, while research was also conducted on the manufacture and reactions of elemental phosphorus. Plans were also announced to extend the field of chemical research and to strengthen the company's scientific staff.

Polystyrene Plant for Greece

The Dow Chemical Co. is to form a wholly-owned subsidiary to manufacture polystyrene plastic in Greece The Greek Government has approved Dow's investment in a plant to cost in excess of a million dollars. It will be built at a location yet to be determined, and construction will begin as soon as a site has been selected. It is expected that operations will start in early 1961.

The plant, which will use imported raw materials, will be the first to produce, polystyrene in Greece. The output is scheduled for both the Greek market and for export.

The company will be held financially by Dow Chemie A.G., of Basle, Switzerland, a wholly-owned subsidiary which manages Dow investments abroad.

Extension to Pernis Plant

A further Shell development following upon the success of Shell's synthetic glycerin in international markets is the decision to increase the capacity of the synthetic glycerin plant at Pernis (Rotterdam), which has been in operation for a year. As a first stage in the expansion programme, the capacity will be increased to a minimum of 15,000 tons a year. The increased capacity is expected to become effective early in 1960.

Chemical Sales Section

British Celanese Ltd. have set up a Chemical Sales Development Section, which, among its other activities, will handle inquiries for chemicals which are or which may become required on an industrial scale, whether or not these are at present in production. Inquiries should be addressed to British Celanese Ltd., Chemical Sales Development, Foleshill Road, Coventry.

Australian Subsidiary for Mono

Since opening their branch office in Australia, developments in the use of their products have led Mono Pumps Ltd. to establish a subsidiary, Mono Pumps (Australia) Pty. Ltd. Chairman of the new subsidiary is Sir Giles Chippendale and managing director is Mr. G. H. Snow, who established the company's first branch office in Australia.

Largest Sulphur Plant

The world's largest plant for the extraction of elemental sulphur from raw natural gas will be built in the Panther River area of Canada's Alberta foothills belt, some 40 miles north-west of Calgary, if present negotiations between the Alberta Pacific Sulphur Co., the Union Oil of California and Canadian Homestead Oils are successfully concluded. The plant will cost some $10 to 12 million to build.

Styrene for Australia

Construction has begun on a styrene plant for C.S.R.C.-Dow Pty. Ltd. at Altona, Australia. George Wimpey & Co. Ltd. have been awarded the contract to build the facilities in association with Bechtel Corp., of San Francisco, and McDonald Constructions Pty. Ltd., of Sydney. The plant is scheduled to come on stream on July 1, 1961.

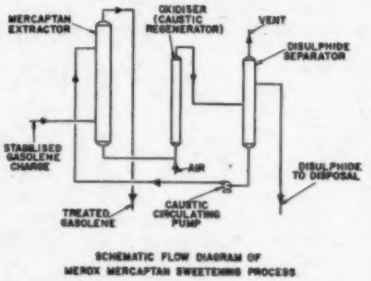
New Refining Process

A new petroleum refining process has been announced by Universal Oil Products Co., of America. The process, Merox, is described as an inexpensive way to remove harmful sulphur compounds from a wide range of gasolene and light distillate oils using a catalyst of undisclosed type. The process can be used for either mercaptan extraction alone, in which it replaces conventional regenerative caustic washing, or sweetening alone, whence it replaces plumbite or copper sweetening. Alternatively, it can perform both functions in a combined operation. The process, which is available for licence to refiners, functions through catalytic oxidation of mercaptans to disulphides in a caustic solution at ambient temperatures, using atmospheric oxygen. Disulphides formed are insoluble in the solution.

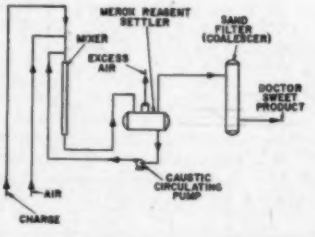
In the application to mercaptan extraction, the Merox process is similar to conventional extraction with caustic solutions, except for the regeneration step which differs markedly. In conventional extraction, regeneration of the caustic is accomplished by steam stripping at the boiling point of the caustic solution. Steam consumption and the corrosive nature of the hot caustic solution make the conventional process expensive. In the Merox process, however, regeneration is accomplished by blowing a small quantity of air through the caustic solution after separating it from the hydrocarbons. Disulphides formed are insoluble in the caustic solution and can be removed by a gravity separator. The caustic is then suitable for reuse. Regeneration is therefore not simply the reversal of the extraction reaction; thus high temperatures and expensive stripping steam are not needed.

In both the sweetening and extraction applications of the process, the catalyst is used either in solution in the caustic or, when greater activity is needed, the catalyst is supported on a carrier and used in a fixed bed. Catalyst life has exceeded 20,000 barrels/lb.

SCHEMATIC FLOW DIAGRAM OF MEROX MERCAPTAN EXTRACTION PROCESS



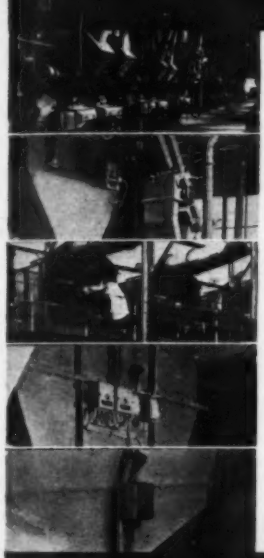
SCHEMATIC FLOW DIAGRAM OF MEROX MERCAPTAN SWEETENING PROCESS



Whatever your level problem

Wherever your level problem

Coal and Coke	Tanks
Sugar	Tankers
Water	Hoppers
Cement	Conveyors
Food, etc.	Silos, etc.



The Fielden *CAPACITANCE METHOD* is the simplest answer

The principle of capacitance control and indication of level for industrial purposes has been successfully used by Fielden Electronics Ltd. for the past ten years. In that time it has produced a range of instruments used throughout the world to answer almost every level problem involving any liquid or free-flowing solid regardless of conductivity, insulating value, density or corrosive properties. Installation is simple, there are no moving parts in the container and direct contact with the material is unnecessary.

Send for fully illustrated literature.

The range includes the famous

TEKTOR Regd.
Level Controller

TELSTOR Regd.
Level Indicator

Fielden

FIRST IN LEVEL

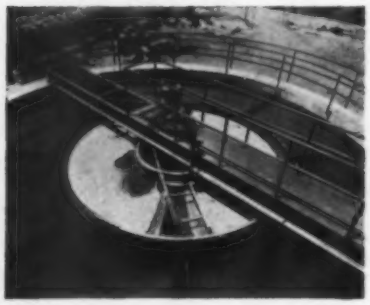
Indicating
Recording
Control

FIELDEN ELECTRONICS LTD · WYTHENSHAWE · MANCHESTER Phone: Wythenshawe 3251 (4 lines) Grams: Humidity Manchester
ALSO AUSTRALIA, ITALY AND CANADA *Branch Offices* LONDON, WALSALL, STOCKTON-ON-TEES, EDINBURGH and DUBLIN
Agents throughout the world
TT2

PLANT EQUIPMENT NEWS

Plant for Treatment of Effluents

An entire range of treatment equipment, based upon well-tried U.S. designs, is now available in the United Kingdom. The range includes clarifiers and thickeners, reactor-clarifiers, flotators, flotator-clarifiers, grit separators, flocculators, filters, and so on. Clarifiers and thickeners can be supplied in sizes up to 325 ft in diameter. Special features available on all machines are spiral inlet diffusers and feed wells of large diameter, features which provide excellent removal of solids and flexible operation.



Flotator-clarifiers of the type shown in the accompanying illustration remove slow-settling solids by flotation, an operation which is performed in the middle of the machine. The heavier solids settle to the bottom, where a raking action moves them to the middle for discharge. This method has the following advantages; it provides a smaller sized basin, prevents odours on account of its rapid removal of solids, and it prevents septic conditions. **Eimco (Gt. Britain) Ltd., Gateshead, Co. Durham.**

BCE 6752 for further information

Bolt-hole Boring Machine

A new machine for boring the holes in coupling flanges especially where space is restricted makes use of a boring bar which is carried in runs in needle-roller bearings fitted in steadying brackets located on each side of the coupling flange. One of these steadying brackets carries a gearbox from which the boring bar is driven, the primary drive being provided from a separate motor by means of a heavy-duty flexible shaft. The gearbox carries, in addition, the feed mechanism which is engaged by means of a thumb screw attached to the feed bracket. The entire assembly is readily clamped to the flanges to be bored and the position of the cutter for the size of hole required is obtained by means of a special micrometer. **Buma Engineering Co. Ltd., Newcastle upon Tyne, 6.**

BCE 6753 for further information

Industrial Oil Burners

Swinney Bros. have designed a completely new range of industrial oil burners of the proportioning low-pressure air type called the "Moduflame" burners. Output is controlled by movement of one lever, either manually or by automatic control, and the air/fuel ratio is maintained constant throughout the operating range. The turn-down range is about 4½ to 1. When using heavy fuel oils which require preheating before burning, the hot oil can be circulated through part of the burner body during non-firing periods. This is said to be a unique feature resulting in trouble-free ignition. **Swinney Bros. Ltd., 34 Crutched Friars, London, E.C.3.**

BCE 6754 for further information

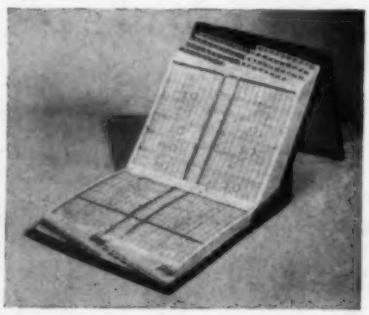
New Range of Worthington-Simpson Pumps

Worthington-Simpson Ltd. announce an addition to their already extensive range of versatile "Monobloc" centrifugal pumps. The new range is designated type "DO" and comprises open impeller-type pumps specially designed to handle slurries, paper stock, liquids containing small solids, and viscous liquids as well as clear liquids. Capacities of from 10 to 360 gpm and heads up to 86 ft, depending on capacity, are covered by this range of pumps. **Worthington-Simpson Ltd., Lowfield Works, Newark, Notts.**

BCE 6755 for further information

Non-mechanical Calculator

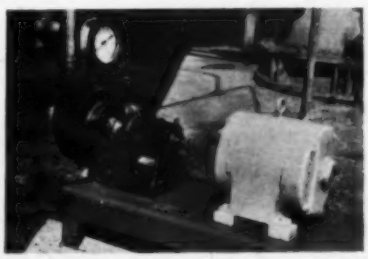
A well-established continental calculator is now being introduced into Britain. It consists of a set of 99 stiff cardboard sheets tabbed with their numbers and bearing precalculated tables which include figures for both direct multiplication and for a number of the most frequently occurring fractions. Up to 9999 can be multiplied by two digits and the answer directly read off the tables. For larger numbers of digits it is necessary to carry out two or possibly more inspections of the tables and to add together the two answers. Thus it is possible to multiply 8 by 2 digits with one addition or 4 by 6 digits with 2 additions. All answers are given directly in (usually 5) digits without any need for estimating intervals and interpolation as in the use of slide rules. The device is called the Bergman Unical Universal Calculator. It weighs 7 lb., so that it can easily be carried around; having no moving parts, it requires no mechanical maintenance. It is manufactured by Unical S.A. of Frauenfeld, Switzerland, costs £25 and the British agent is **Kenneth Miles, 13 Netherall Gardens, London, N.W.3.**



BCE 6756 for further information

Corrosion-resistant Mechanical Seal

This seal proved an excellent solution to an awkward pumping problem involving the circulation of 98% sulphuric acid around scrubbing towers. For this purpose, stainless-steel pumps were employed each handling about 18 Imperial gpm at an 80-ft head. Corrosion of the seal components and wear of the seal faces were the main troubles experienced before the adoption of the new seals, and attention, adjustment and replacement were frequent with the original seals. Normal operating temperatures were in the range 35-70°C.



The replacement seal used above, known as the Chemiseal, is a chemically resistant, pressure-balanced unit with spring-loaded PTFE bellows. It can operate without trouble at 75°C and at 100 psi, and at shaft speeds up to 2000 fpm without cooling. In the plant referred to, more than two years elapsed before the replacement seals required attention of any kind. **The Garlock Packing Co., Palmyra, New York.**

BCE 6757 for further information

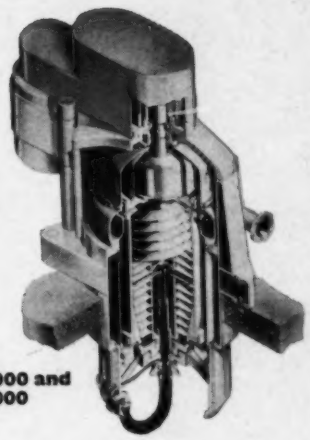
Neutralisation Plant

It has been generally accepted that cyanide- and chromate-bearing effluents are extremely difficult to treat automatically, but the use of a newly developed neutralisation plant makes it possible for any company to have an automatically controlled effluent of *pH* within the range 6.5-7.5. This is a figure generally acceptable to Water and River Boards throughout the country.

In this treatment unit, developed by A. M. Lock & Co. Ltd., the effluent is

The world's biggest range of scroll discharge centrifuges

Sharples Super-D-Canters



P.4000 and P.7000

The Sharples range of continuous centrifuges not only includes machines of every type but a full range of models of each type.

Take for instance the Super-D-Canter type scroll discharge centrifuge which is widely used for centrifugal filtration of crystalline and amorphous solids from slurries. There are no less than 8 basic models.

P.400 Laboratory Scale Super-D-Canter—4″ diameter conical bowl; maximum centrifugal force 2,100 x G.

P.600 A new Pilot Plant Scale Super-D-Canter—6″ diameter cylindrical/conical bowl; maximum centrifugal force 3,150 x G.

P.1000 Full Scale Super-D-Canter—14″ diameter conical bowl; maximum centrifugal force 2,100 x G; especially suitable for handling crystalline materials where good washing is required.

P.2000 Full Scale Super-D-Canter—14″ diameter cylindrical bowl; maximum centrifugal force 2,100 x G.

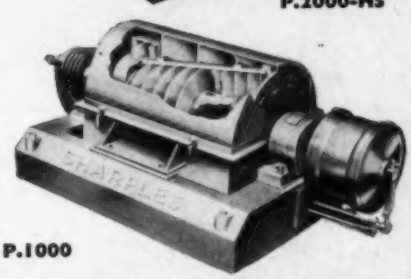
P.2000-HS High Clarifying Capacity Super-D-Canter; 14″ cylindrical/conical bowl; maximum centrifugal force 3,200 x G; especially suitable for handling fine solids.

P.3000 A new high liquid throughput Super-D-Canter; cylindrical/conical bowl; maximum centrifugal force 3,200 x G; especially suitable where high throughput of slurries at relatively low solids content have to be handled.

P.4000 A new vertical type Super-D-Canter; cylindrical/conical bowl; maximum centrifugal force 3,200 x G; especially suitable for pressurised operation up to 150 p.s.i.g. at high liquid throughputs.

P.7000 Another new vertical Super-D-Canter suitable for very large solids throughput; maximum centrifugal force 1,700 x G; available for pressurised operation up to 150 p.s.i.g.

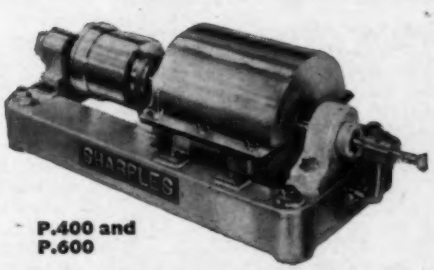
For each of these models there are detailed specifications for condition of operation such as for vacuum or pressure, for low temperature, for special rinse requirements, for construction in various materials, etc.



P.2000 and P.2000-HS

P.1000

P.3000



P.400 and P.600

SHARPLES

SHARPLES CENTRIFUGES LIMITED

TOWER WORKS, DOMAN ROAD, CAMBERLEY, SURREY

Phone. Camberley 2601 *Grams: Superspin, Camberley*

pumped through a continuous flow detector assembly constructed from inert plastic material and fitted with detector electrodes. The cyanide detector assembly generates a potential depending mainly upon the concentration of cyanide present, and a high potential is set up in the chromate detector when chromate in the hexavalent state is present; and a potential of only few millivolts when hexavalent chromate is reduced to the trivalent state. The potentials are electronically amplified and changed into current outputs of 0-1 mA which are fed into a reaction controller.

Fully-automatic *p*H control assemblies incorporating continuous flow or immersion-type electrode systems, *p*H amplifiers, electronic dose rate controllers and reagent control valves are provided to maintain the *p*H above 10.5 for cyanide treatment or below 3 for chromate reduction and for the automatic neutralisation before discharge to drains and rivers. The neutralisation plant can be installed for cyanide or chromate treatment individually or, where necessary, both effluents can be treated at the same time. **A. M. Lock & Co. Ltd., Oldham.**

BCE 6758 for further information

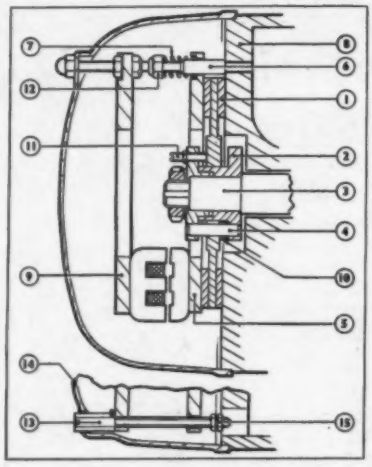
P.T.F.E./Glass Stopcock

A small valve which will satisfy a long-felt need in laboratory and pilot plant is now available from the Loughborough Glass Co. The barrel and side arms are made from Pyrex glass and the key is made from P.T.F.E. This latter component requires no lubrication and will not seize even in the presence of strongly alkaline solutions. The surface of the key has dimples which deform when the retaining cap is tightened so that an excellent seal is obtained suitable for medium vacuum work. The bore of the cock is 3 mm. **Loughborough Glass Co. Ltd., Loughborough, Leicestershire.**

BCE 6759 for further information

Magnetic Brakes

Newman Industries have extended the range of magnetic brakes which can be built into their three-phase motors.



The legend to the annotated diagram of the Newman magnetic brake is as follows: (1) *brake disc;* (2) *steel hub;* (3) *motor shaft extension;* (4) *steel driving pins;* (5) *armature;* (6) *guide pins;* (7) *disc clamping springs;* (8) *non-drive end bracket;* (9) *cast iron plate magnet;* (10) *springs;* (11) *set screws;* (12) *adjusting locknut;* (13) *manual release screws;* (14) *cover; and* (15) *locknuts.*

Eventually the brakes will be supplied in a torque range of 3-120 lb./ft covering a motor horse-power range of fractional up to 40. On motors above 3 hp alternative brakes can be provided, one having a retarding torque just in excess of the full motor torque and the other 1½ times the full motor torque. The brakes are extremely rapid in action and are of the fail-to-safe type. **Newman Industries Ltd., Yate, Bristol.**

BCE 6760 for further information

Dual-mould Automatic Injection Moulding Machines

A recent design of injection-moulding machine is provided with two moulds. It is therefore capable of achieving a higher output without increasing the overall size and price of the machine or the complexities and expense of the moulds. Design is based on the fact that when moulding plastic articles, the comparatively high proportion of the total cycle time used for injection and ejection can be used for cooling if two moulds are available and thus make more intensive and efficient use of the facilities provided on the machine. Yet such a machine must remain comparatively simple in construction and avoid the shortcomings of rotary multi-stage machines which suffer from unnecessary complexity and high mould costs. The basic principle of operation of the Polymat machines is the use of "Dual Moulds", i.e., two identical or similar moulds are mounted on a mould plate which rotates 180° round a centre pivot in alternating directions at every cycle. Thus, while one mould is placed against the injection nozzle, the other is being cooled on the opposite side of the mould plate. The finished moulding is ejected before the cooled mould returns to the filling position for its next charge. The entire cycle is automatic and continuous. **John Kimbell & Co. Ltd., 4 South Lambeth Place, London, S.W.4.**

BCE 6761 for further information

Pump for Corrosive Liquids

The main part of this pump is a rotor with longitudinal grooves into each of which fits a free roller. The combined action of centrifugal force and liquid pressure forces the rolls against the walls of the cylindrical pump barrel which is eccentric with respect to the rotor. The inlet and outlet connections are situated at the sides of the cylindrical housing. The pump and its parts can be supplied in a range of materials. Thus the rotor can be supplied in Ni-resist, the shaft in 18/8 stainless steel, the housing and connections in hard chromium-plated pearlite and the rollers in rubber or nylon. Two sizes of pump are available, one with a capacity of 250 l. per min. and a delivery pressure of 15 atm. and the other with an output of 90 l. per min. for a pressure of 25 atm. **Lindinger Agro Co. A.S., Bulow Agro Co., Copenhagen, Denmark.**

BCE 6762 for further information

Non-electric Magnetic Drum Separator

A separator which meets all the requirements of the modern heavy media recovery flow sheet has among its advantages the elimination of densifier. This demands a magnetic concentrate of at least 2.2 sp. gr. when using magnetite, and the desirability of treating the effluent from the wash screens directly without pre-thickening makes it essential to achieve the maximum magnetite recovery even with very dilute slurries.

The 30-in. dia. drum, provided with expendable stainless-steel outer covers, and available in widths up to 72 in., is energised by a powerful multipolar permanent magnet unit. Single-stage recovery at a feed rate of 80 gpm per foot of magnet is exceptionally high under a very wide range of feed concentrations, and recoveries as high as 99.6% at 1.38% magnetite in feed (by weight) have been obtained. This represents a magnetite consumption of less than a tenth of a pound per ton of coal washed, based on an average of 150 gal. of wash water per ton of coal. A clean magnetite concentrate of 2.4 sp. gr. is produced, independent of feed concentration. **Rapid Magnetic Machines Ltd., Lombard Street, Birmingham, 12.**

BCE 6763 for further information

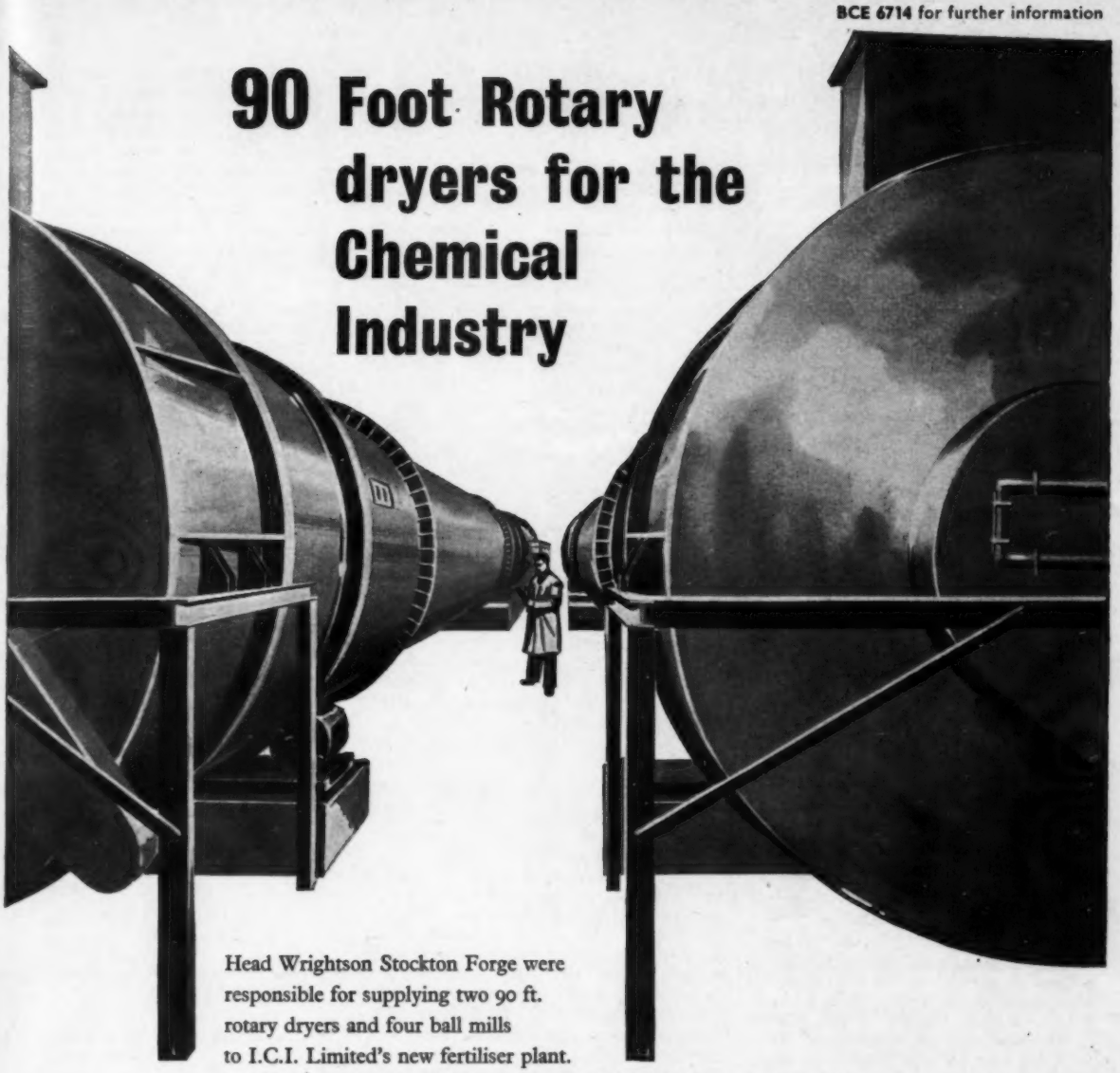
Crusher Control

By means of two electronic relay units it is possible to stop an electric motor when it becomes dangerously overloaded and to start it again once normal operating conditions have been restored.

A typical application is a conveyor-fed crusher which is to be protected from overloading. By means of an over-

BCE 6714 for further information

90 Foot Rotary dryers for the Chemical Industry



Head Wrightson Stockton Forge were responsible for supplying two 90 ft. rotary dryers and four ball mills to I.C.I. Limited's new fertiliser plant. The dryers are among the largest machines of this type ever to be built.

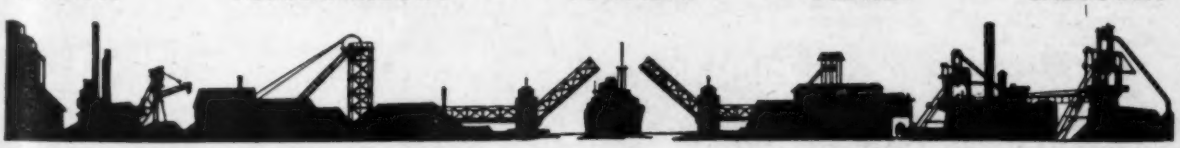
Head Wrightson have specialised in the design and manufacture of drying and grinding equipment for the Chemical and Mining industries for many years and have installed similar equipment in many plants in this country and abroad.

Please write for further information or telephone Stockton 65268.

HEAD WRIGHTSON STOCKTON FORGE LTD

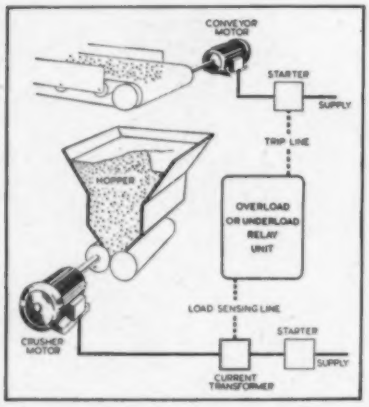
STOCKTON FORGE WORKS · STOCKTON-ON-TEES

LONDON · JOHANNESBURG · TORONTO · SYDNEY · CALCUTTA



P.3792

PLANT EQUIPMENT NEWS



load relay in the motor circuit, the conveyor drive can be stopped before the crusher motor approaches an overloaded condition. The conveyor motor can be made to start up again when the load upon the crusher motor falls to a safe value.

The units in combination can also be used as a direct on-off control and by linking the overload relay with a single-stage timer, an automatic restart can be made after the elapse of a set time interval. Alternatively, a cyclic timer can be used if a succession of restarting attempts is the normal requirement. **Elcontrol Ltd., Hitchin, Hertfordshire.**

BCE 6764 for further information

Intensive Mixer

A range of double rotor intensive mixers for high-speed uniform mixing of all types of plastic and rubber compounds was introduced recently by Fawcett, Preston & Co. Known as the Bolling Spiral-Flow, they are particularly suited to polythene and butyl rubber where high temperatures are necessary. By employing flood lubrication temperatures up to 400°F can be permitted. The mixer is made in a range of five sizes starting with the size No. "0", with a batch capacity of 11 lb. of 1.50 specific gravity stock. This model is designed for laboratory or small production work. At the opposite end of the range is size No. 12, with a batch capacity of 600 lb. of 1.50 specific gravity stock. This machine requires a motor of hp in the range 450-750. They all have spiral flow sides heated or chilled through the same passages, accessible for cleaning from outside. The rotors are of best-grade cast steel with Stellited flights and full-circle shroud flanges. The mixers are complete with all necessary control equipment for air, steam and water services. Various rotor speeds are available to suit each particular application. **Fawcett, Preston & Co. Ltd., Bromborough, Cheshire.**

BCE 6765 for further information

Unit-type Shaft Seals

To meet the need for simple and inexpensive shaft seals for small pumps for low-pressure duties, the Morgan Crucible Co. have recently introduced two unit-type seals. The first, known as the Unit seal, consists of a carbon ring bonded to a synthetic oil- and heat-resisting rubber bellows enclosing a stainless-steel spring. For this design there are detail modifications. The second seal, the Positive-drive unit seal, is similar, except that in this case the assembly is an easy push-fit into a pressed brass driving cap. This latter has indentations into which the driving lugs of the carbon ring fit when the seal is compressed to the full extent of its working length.

The advantages claimed for these seals are low initial cost, ease of fitting, trouble-free operation at all speeds, long life and compactness. In addition, they are unaffected by dirty liquids, they will tolerate temperatures as high as 100°C, and they will seal effectively against pressures in either direction; normally the pressures up to 10 psi differential can be contained, but higher pressures also can be accommodated. **The Morgan Crucible Co. Ltd., Battersea Church Road, London, S.W.1.**

BCE 6766 for further information

Carbon Vanes for Pumps

A range of self-lubricating carbon vanes obtainable from the Morgan Crucible Co. is suitable for sliding-vane type pumps and compressors, especially where the service conditions are damp and corrosive. Since these vanes are self-lubricating, they are also particularly suitable where air or gases have to be handled without becoming contaminated by oil. Three grades of carbon are available, the choice for a particular application depending upon the operating conditions, such as the gas or liquid to be handled, the peripheral speed, type of rotor, differential pressure, and whether radial or inclined slots are used in the pump or compressor. **The Morgan Crucible Co. Ltd., Battersea Church Road, London, S.W.11.**

BCE 6767 for further information

Accurate Temperature Controller

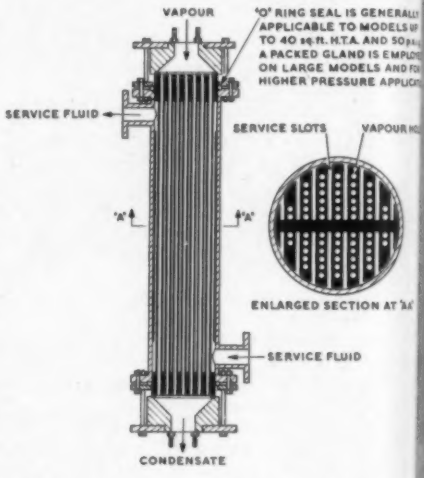
Claimed to be the most accurate temperature controller commercially available, a new platinum-resistant thermometer has an accuracy of ±0.025°C at 1000°C. This high degree of accuracy can be maintained for months and years continuously. The instrument is suitable for the close control of processing equipment operating at temperatures up to 1200°C. The thermometer circuit consists of an a.c.-powered Wheatstone bridge, the pre-set arm of which consists of a 1000-ohm potentiometer graded in 4000 steps, each being 0.25°C. The out-of-balance voltage from the bridge is amplified by a two-stage amplifier giving a gain of approximately 2000. The degree of out of balance, that is, temperature error, is displayed by a "magic eye" indicator having two degrees of sensitivity.

The instrument is readily applied to automatic control systems. **Winston Electronics Ltd., Shepperton, Middlesex.**

BCE 6768 for further information

Condenser for Corrosive Vapours

Exceptional resistance to corrosion is offered by a new design of condenser which utilises a graphite heat-exchange element. This condenser avoids the tubular type of construction, since the vapour condenses in the core holes of the element and the coolant, which is the shell-side fluid, flows through the slotted outer surface. The header boxes are also constructed from graphite and the shell, normally of mild steel, can be provided with a resistant lining if a corrosive fluid is used for cooling purposes. The heat-transfer performance is good and the rates that can be obtained are three to four times greater than those obtained with a comparable glass exchanger. The overall condensing coefficient obtained in the case of isopropyl alcohol with these condensers is 160 Chu/sq. ft °C hr. Eight sizes of condenser are available with heat-transfer surfaces ranging from 4 to 200 sq. ft. **Powell Duffryn Ltd., 8 Great Tower Street, London, E.C.3.**


VAPOUR
'O' RING SEAL IS GENERALLY APPLICABLE TO MODELS UP TO 40 sq. ft. H.T.A. AND 50 p.s.i. A PACKED GLAND IS EMPLOYED ON LARGE MODELS AND FOR HIGHER PRESSURE APPLICATIONS
SERVICE FLUID
SERVICE SLOTS
VAPOUR
'A'
'A'
ENLARGED SECTION AT 'A'
SERVICE FLUID
CONDENSATE

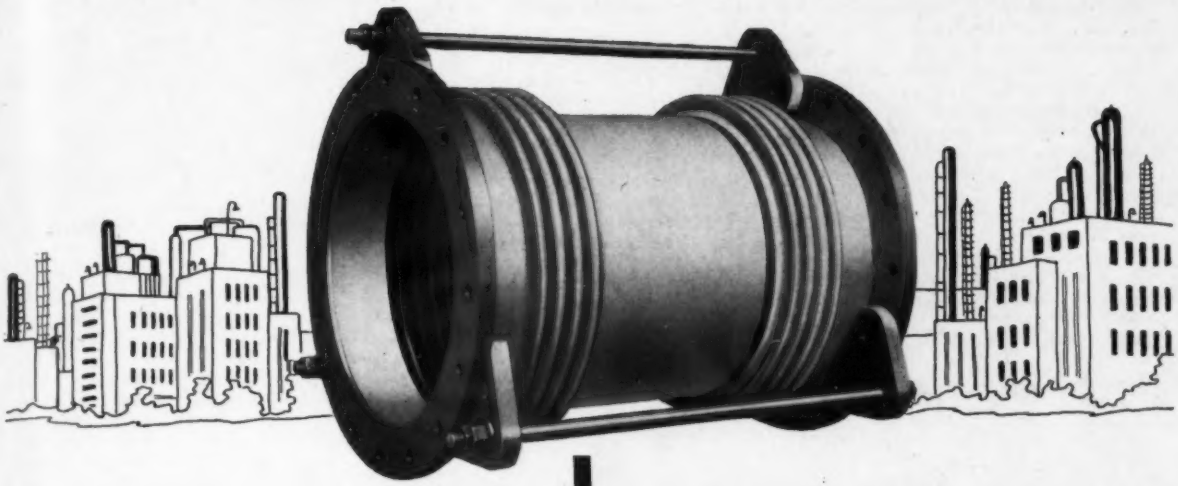
BEC 6769 for further information

Miniaturised pH Meter

The increase in the number of instruments required in modern process control has led to the development of a whole

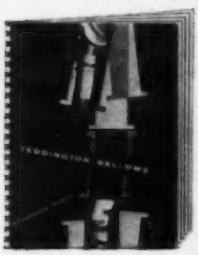
BCE 6715 for further information

The use of **BELLOWS** in Chemical Engineering



Wherever pipes operate under conditions of temperature and pressure, expansion joints may be required. The bellows type expansion joint has in the past few years become standard equipment, being now recognised as extremely efficient. Teddington engineers have played a leading part in the introduction of stainless steel bellows expansion joints to the chemical industry and can be relied on to co-operate fully with design and plant engineers who are confronted with pipe problems.

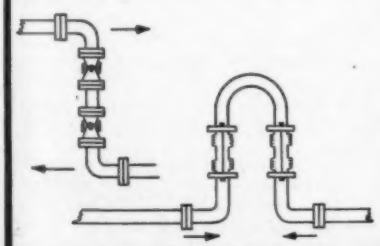
SEND FOR HANDBOOK No. M 12



Illustrated descriptions are given of the many and varied problems that Teddington Bellows Expansion Joints can overcome. If you did not secure your copy at the Engineering Exhibition, please write for one.

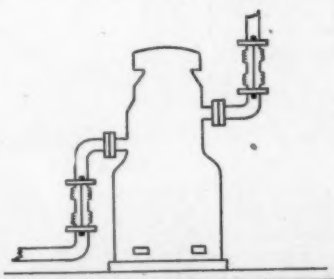
IN PIPELINES

Restrained double units operate in shear to accommodate large movements with negligible anchor leads. Two restrained double units can be used to make a very flexible, small dimension pipe loop for large expansion.



ISOLATING PUMP MOVEMENT

In the pipework connected to a pump, restrained double units fitted at right-angles to the plane of vibration will operate in shear to accommodate the movement and protect the pipework.



SEALING VALVE STEMS

Bellows units provide an absolutely leak-proof seal on valve stems.

TEDDINGTON

STAINLESS STEEL FLEXIBLE BELLOWS EXPANSION JOINTS

TEDDINGTON AIRCRAFT CONTROLS LTD (*Industrial Bellows Division*) Ammanford, Carmarthenshire. Tel: Ammanford 2255

PLANT EQUIPMENT NEWS

range of miniaturised instruments for measuring temperature pressure, level and flow. The first industrial *p*H meter in this form is announced by Electronic Instruments. This industrial *p*H meter occupies 60% less panel area and may be used in conjunction with any of the conventional miniaturised recorders or controllers. The performance of the miniaturised *p*H meter is identical to that of the standard instrument and servicing may be carried out by simply opening the glass-fronted door and withdrawing the entire chassis. **Electronic Instruments Ltd., Lower Mortlake Road, Richmond, Surrey.**

BCE 6770 for further information

3000-lb. Rating Hydraulic Four-way Valve

A new range of pneumatically-operated four-way valves for the control of double-acting hydraulic rams eliminates the need for two three-way valves of the conventional type. These valves, which are suitable for operation either through a manual pilot or an electric timer, are operated by a single diaphragm top. This simplification is achieved by using the line pressure to operate one section of the valve on a differential area principle. A two-way valve may also be installed in one of the cylinder lines to allow the ram to be stopped in any position desired. The water hammer or shock due to sudden release of pressure in the valve is eliminated by the use of V-slots in both inlet and drain seat rings of the valves. The valves are suitable for hydraulic working pressures from 200 psi up to 3000 psi (non-shock) at temperatures up to 140°F, and are made in sizes ¼ in., 1 in., 1½ in. and 2 in. A 300-lb rating four-way valve is also available in sizes ¾ in. and 1¼ in. only. **Hunt & Mitton Ltd., Ooozells Street North, Birmingham, 1.**

BCE 6771 for further information

Rotary Drier and Cooler

A unit designed originally for the drying of sand but with obvious potentialities for the drying of other granular materials is now available from Modern Furnaces & Stoves. It consists of a rotary drum equipped with flights in which is fitted a rotating gas- or oil-fired heat exchanger. The material to be dried comes into contact with the hot surface of the exchanger which rotates with the drum, and the material is therefore treated without coming into contact with the products of combustion.

The drier is fed by means of a hopper and by a system of buckets arranged on the periphery of the drum. This latter directs the material into the annular space between the drum and heat exchanger. From the drying section the material is conveyed by the action of the flights into the cooling section, where the material falls through an induced draught of cold air. Entrained solids are recovered from the air stream by means of a cyclone connected to the discharge of the induced-draught fan.

The entire unit complete with burner is mounted upon a substantial bedframe provided with elevating screws for adjusting the tilt of the drum. The rate of passage of material through the unit can therefore be varied. Two sizes are available, one capable of a throughput of one and the other of two tons per hour for a feed containing 5% moisture. Fuel consumption for the complete drying of a granular product with this amount of moisture is around 500 cu. ft of gas or 1.3 gal. of oil per ton of dry product. **Modern Furnaces & Stores Ltd., Booth Street, Handsworth, Birmingham, 21.**

BCE 6772 for further information

Slide Rule for Calculating Water Distribution

A slide rule with a vertical attachment has been invented in Israel. It is claimed to speed up considerably calculation of water distribution and flow in pipelines. Before this, water flow in networks were computed either on an electric analyser or by a tedious procedure based on trial and error calculations of flow and distribution in pipelines that are composed of various pipe diameters and lengths. Calculations of this type are constantly required, not only in determining the behaviour of water in future lines, but in extensions and repairs on existing ones. By a vertical as well as horizontal attachment on the rule, all diverse factors of diameters, lengths, velocities and types of pipe appear each as a unit of measure, and the rate of flow of water through the various sections can immediately be determined. In converse, given amounts of water can be planned to flow by the introduction of various pipes and fittings in accordance with the answers which the slide rule records. **Technion, Institute of Technology, Israel.**

BCE 6773 for further information

Flow Sheets for Contact Sulphuric Acid Plants

More than 750 variations of sulphur-burning contact sulphuric acid plants are possible. Consequently, the would-be purchaser of such a plant has a wide field to choose from. To facilitate the task of selection the Chemical Construction (G.B.) Ltd. have produced a brochure which includes a pictorial flow sheet to which are attached a series of flaps. Each of these contains a flowsheet alternative. For example, raising the flap depicting a fire-tube waste-heat boiler downstream of the sulphur burner reveals, as an alternative, a forced-circulation type of boiler which is to be preferred for plants of approx. 200 tpd output or more.

Each major item of plant is treated in this way, and is also provided with a description of its function. Where an alternative is available, the reason for its use is given.

The use of colour is effective. The basic plant is indicated by means of brown numerals, and the alternative equipment is shown by blue numerals. A check card is provided upon which the sulphuric acid manufacturer can record his choice of plant. In addition to its value in buyer-purchaser relations, this brochure should prove extremely useful to students and teachers of chemical engineering, since it is an excellent means of demonstrating the diversity of sulphuric acid plants. **The Chemical Construction (G.B.) Ltd., 9 Henrietta Place, London, W.1.**

BCE 6774 for further information

Flame-proof Electronic Gauge

A new electronic tank gauge for use on deep storage tanks is obtainable from Firth Cleveland Instruments Ltd. The new model is housed in a ductile iron case and is flameproof. This new Gilbarco gauge retains all the features of the older aluminium-cased model, including accuracy in the field to $\frac{1}{16}$ in. and in the laboratory to $\frac{1}{64}$ in. The gauge reads the true level of the liquid in ft, in. and $\frac{1}{16}$ in., regardless of its specific gravity. It is power-actuated to overcome the effect of varying friction in pulley sheaves and the variation in the weight of the measuring element at high and low liquid levels. It can provide the data necessary for a co-ordinated system of remote transmittion. The gauge can be equipped for automatic selection of temperature-averaging resistance thermometers, and can be fitted with high- and low-level alarms.

Corrosion-resistant materials are used in its construction and it can be installed, serviced, re-calibrated and maintained without taking the tank out of service. The gauge will also measure the interface between two immiscible liquids. In large tank farms the gauge on each tank can be connected to a null-balance receiver for remote liquid level and average temperature readings. The standard receiver is equipped to read this data from 30 tanks. A means of remotely checking and proving the accurate operation of each gauge from the instrument house is incorporated in the receiver. An air-purged version of this can be supplied for

So simple!

So surely SIMPLIFIX the foolproof coupling!

Simplifix couplings form a perfect joint on almost any kind of tubing including those with very thin walls. All that is required is tightening with a spanner – no special work is required on the tube and the anti-friction washer prevents the tube twisting when the nut is tightened. Simplifix couplings are suitable for all pipe line systems up to 2" o.d. In a wide range of interchangeable standard fittings. Non-standard fittings of all kinds can also be made to order. Write for further information and fully illustrated catalogue.

SIMPLIFIX

SIMPLIFIX COUPLINGS LIMITED
HARGRAVE ROAD · MAIDENHEAD
BERKS · TEL: MAIDENHEAD 5100
A member of the ALENCO Group of Companies

PLANT EQUIPMENT NEWS

mounting in hazardous areas. **Firth Cleveland Instruments Ltd., Stornoway House, Cleveland Row, St. James's, London, S.W.1.**

BCE 6775 for further information

Interchangeable Argon and Flame Ionisation Detectors

New argon and flame ionisation detectors have been introduced by Shandon as an important addition to their Universal Gas Chromatograph. These two new detectors, and the new high-efficiency amplifier designed for use with them, are readily interchangeable with the flame detector in existing equipment. The argon detector is also an extremely sensitive detector with applications similar to the flame ionisation unit. The carrier gas is argon, which emerges from the column into a weak radioactive field in the detector. A potential of about 2000 V, supplied from batteries, is applied at the electrode. Ionisation of the fractions as they are eluted causes a variation of current. The resultant signal is fed to the same amplifier and control unit as is used with the flame ionisation detector.

A versatile new amplifier enables users of the equipment to obtain the maximum possible value from the sensitivity of flame ionisation and argon detectors. The high sensitivity of flame ionisation and argon detectors allows full advantage to be taken of the efficiencies—up to 30,000 theoretical plates — obtainable from a 50-ft column. Furthermore, the instrument may now be fitted with any available type of sample-injection system for both liquid and gas samples. In general, the makers claim that the unique modular design allows for all foreseeable developments to be incorporated at minimum cost. **Shandon Scientific Co. Ltd., 6 Cromwell Place, London, S.W.7.**

BCE 6776 for further information

Refrigeration Units for Air Conditioning

In a new unit use of a "packaged" compressor is claimed to make the refrigerating set one of the most up-to-date available. It is a high-duty refrigeration unit specially designed for air-conditioning installations in large buildings but usable for refrigeration plants in chemical works, breweries and cold stores of all kinds. The refrigerant used is Freon 12 and the high efficiency of the turbo-compressor is enhanced by interstage flash cooling. Capacity is between 1,500,000 Btu/hr (125 tons) and 9,500,000 Btu/hr (800 tons). The set comprises the turbo-compressors, condenser and evaporator, ready assembled for ease of erection on site. It is said to be quiet, vibration-free and of low weight, giving high plant efficiency as a result of multi-stage compression of the Freon gases and two-stage expansion. **Sulzer Bros. (London) Ltd., 31 Bedford Square, London, W.C.1.**

BCE 6777 for further information

New Publications

"The Prevention of Fire." This gives details of the work of the Fire Protection Association in respect of their Information Centre, Standard Recommendations and Advisory Publications. **Fire Protection Association, 31/45 Gresham Street, London, E.C.2.**

BCE 6778 for further information

Protective gloves for industry have to be chosen with great care. They must provide complete protection under the conditions of the work being performed, and they must be "usable" for a particular job, otherwise safety and protection will not go hand in hand if the glove hinders the operation of the worker.

Modern industry presents many problems in combining these two essential qualities in one glove and, consequently, a very wide range of carefully designed gloves is needed to fit in with all industrial demands. Protective gloves for many industries has been the subject of special study for many years by Waddington's of Hull, who have developed a wide range of safety gloves which includes several hundred different types. Full details are given in Waddington's handbook of "Furno" industrial gloves and protective clothing. **G. Waddington & Son Ltd., Newland, Hull, Yorks.**

BCE 6779 for further information

"Plastics Serve Everyman." A 12-page booklet for the general reader. **Distillers Plastics Group, Piccadilly, London W.1.**

BCE 6780 for further information

Special Advanced Courses Northern Region Session, 1959-60. The courses shown cover both technology and commerce. The majority are specialised and designed for those at or about graduate level, but a few others of a non-specialised character have been included as being of value to industrial and commercial personnel. **Special Advanced Courses Sub-Committee, Northern Advisory Council for Further Education. Grosvenor Road, Newcastle upon Tyne, 2.**

BCE 6781 for further information

Bulletin of Advanced and Special Courses. This lists the same types of course as the above but for a different area. **Regional Advisory Council for the Organisation of Further Education in the East Midlands, Bridlesmith Gate, Nottingham.**

BCE 6782 for further information

"Training Services in Industry." In this booklet is an account of the development of training schemes and advisory services. It describes the main functions of these services under the headings "Provision of information", "Practical training and training aids", "Supervision of training and further education", "Recruitment", "Selection and induction", "Indenture and registration", "Tests and certificates" and "Research into training methods". Descriptions are given of the services which have been set up in the iron and steel, engineering, chemicals and other industries. For comparative purposes a brief summary of the training arrangements in the nationalised industries is included. Services and assistance also available to employers from the Ministry of Labour and a number of voluntary bodies find a place in the booklet. **Industrial Training Council, Smith Square, London, S.W.1.**

BCE 6783 for further information

A brochure, "Plastic Coating, Spraying, Dipping, Casting", illustrating a few of the wide range of goods sprayed, dipped and slush cast in a new modern factory. Plastic coating, by combining the properties of plastics with the rigidity and strength of metal, offers a new design approach described in the publication. **Durable Plastics Ltd., Woodbridge Industrial Estate, By-Pass, Guildford, Surrey, England.**

BCE 6784 for further information

"Neumo" pumps can be supplied to pump anything that flows and all pumps are variable in delivery from their maximum to nil. They are air or steam operated and safe for use with low-flash point liquids and for use in flame-proofed areas. In the motor a dead position has been completely avoided and, no matter the position in which the valve or piston comes to rest, the motor will in all cases function normally immediately air is supplied. The brochure, "The Neumo Range of Pumps, Their Basic Principles and Design", is obtainable from **Neumo Ltd., South Coast Road, Peacehaven, Sussex.**

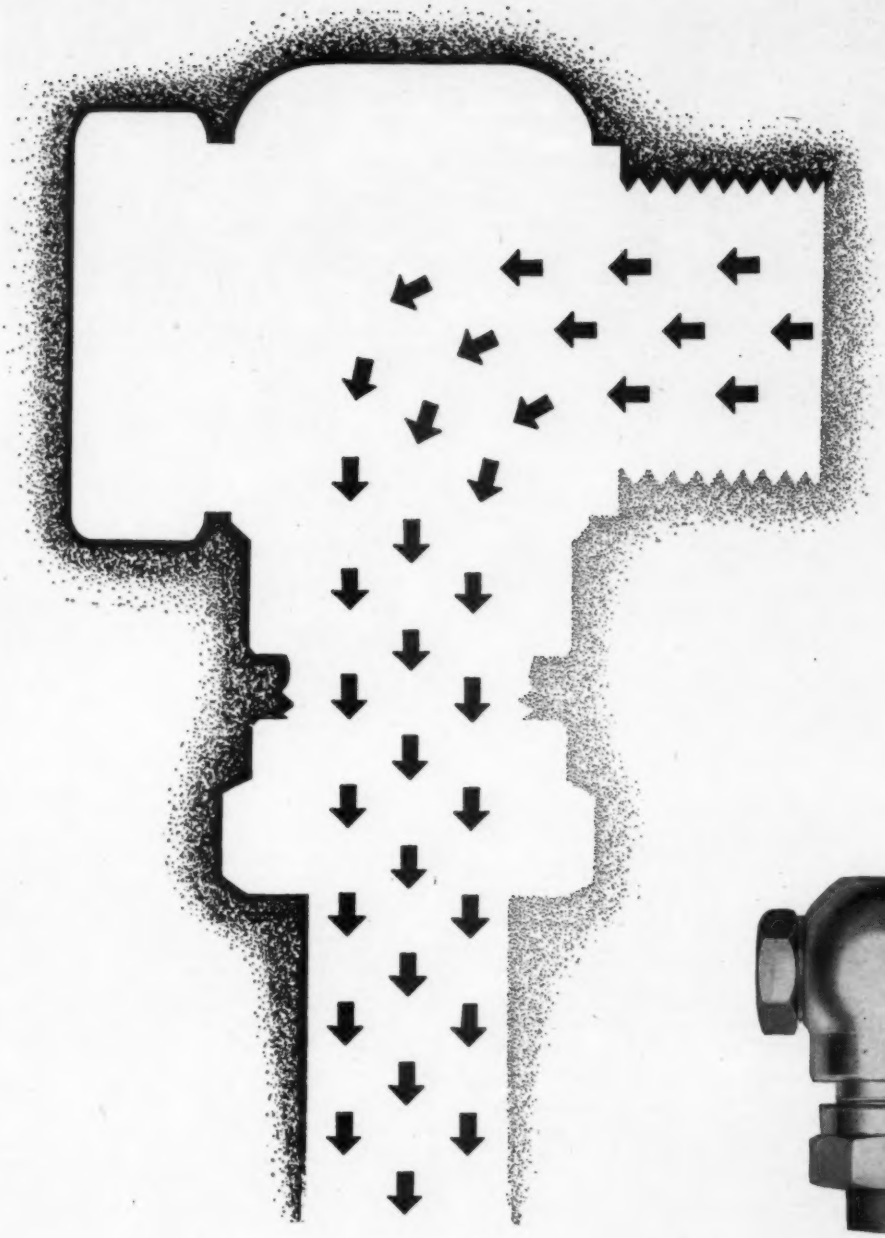
BCE 6785 for further information

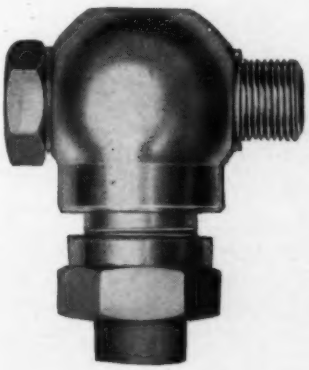
Bulletin of Special Short Courses (including special short courses in higher technology), 1959-60. **Regional Council for Further Education for the South-West, 12 Lower Castle Street, Bristol, 1.** This Bulletin provides advance notice of courses to be offered in the Region during session 1959-60. Many are of a post-graduate nature, but others, although not of that level, are equally worthy of attention.

BCE 6786 for further information

Further Education (Art. Commerce and Technical courses) in the South-West, 1959-60. This booklet summarises the facilities for vocational education and training in the south-west and gives courses classified into: (i) full-time, (ii) full-time "sandwich" courses, (iii) part-time day or day and evening courses, and (iv) part-time evening courses. There is also a detailed index. **Regional Council for Further Education for the South-west, 12 Lower Castle Street, Bristol, 1.**

BCE 6787 for further information





in connection with high pressure

ERMETO HIGH PRESSURE COUPLINGS are specified by many leading manufacturers, because they are the most reliable fittings on the market for every high pressure installation. Our catalogue illustrating the full range of standard fittings will be sent to you on request. Non-standard fittings can also be made to suit your specification. Our technical experience is at your service.

ERMETO **BRITISH ERMETO CORPORATION LTD**

HARGRAVE ROAD • MAIDENHEAD • BERKS • TELEPHONE: MAIDENHEAD 5100 • A member of the ALENCO Group of Companies

Month's News in Brief

First Hydrorefining Plant Contracts Placed

Port Talbot Chemical Co. Ltd., formed jointly by the Steel Company of Wales Ltd. and Lincolnshire Chemical Co. Ltd., have now placed contracts for the first hydrorefining plant in Great Britain to refine coke-oven crude benzole to pure benzene. The plant is designed to treat 4.5 million gal. of crude per annum and provision has been made for doubling capacity. The plant uses coke-oven gas as the source of hydrogen. The hydrorefining plant itself is being supplied by Lurgi, of Frankfurt, Germany, with Simon-Carves Ltd. responsible for erection, the procurement of English supply equipment and the installation of electrical equipment and instrumentation. The new plant is programmed to start up towards the end of 1960, and Cremer & Warner have been appointed consulting engineers for the project.

Hydrorefining of crude benzole has become almost standard practice in Germany, Japan and many continental countries where low-sulphur benzene is in demand, and the unit will be capable of meeting the increasing demand for low-sulphur benzene in this country. In addition, the process overcomes the troublesome effluent problem and high physical losses associated with conventional acid refining. The distillation equipment incorporating the Montz plate has been designed in conjunction with R. & J. Dempster Ltd. to produce maximum yields of 5.35 crystallising-point pure benzene on a continuous unit. Low-sulphur nitration toluene, xylene and special high flash naphtha with improved acid wash test and colour stability will be produced in a vacuum batch column and still. The new works will be within the Port Talbot works of The Steel Company of Wales, whose coke ovens will supply the crude benzole and coke-oven gas. The necessary storage and ancillary plant and an administration and laboratory block will make the works fully self-contained.

A view of the carbonylation alcohol plant operated by I.C.I.'s Heavy Organic Chemicals Division at Billingham, Co. Durham.

U.K.'s Largest Alloy Spun Ends

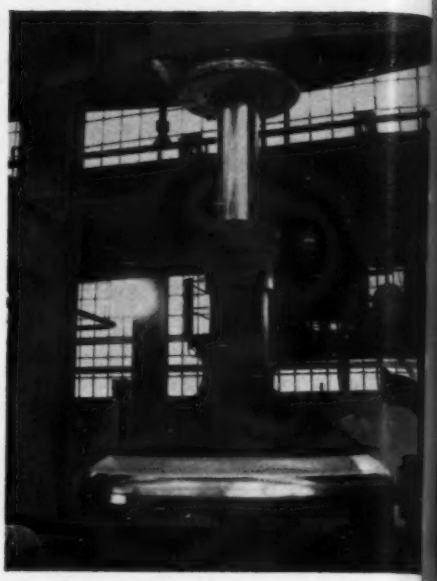
G. A. Harvey & Co. Ltd. have recently delivered to a well-known fabricating company what are believed to be the largest spun aluminium ends ever produced in the United Kingdom. These "Rotarprest" aluminium alloy ends have an inside diameter of 10 ft and are 1¼ in. thick. Each end was spun from two plates (BA. 28-N.P.5/6) and butt welded to form a flat disc 12 ft in diameter.

Special methods of fabricating, developed by the company's Research Department, had to be used in the manufacture of the ends to prevent scoring or galling from the steel tools used in the spinning process. Furthermore, as the maximum temperature of the particular aluminium alloy used for the ends is critical, G. A. Harvey, who previously were used only to handling all types of steels, stainless steels and high nickel alloys, had also to develop a technique of heating the aluminium alloy plates at the normal furnaces used for heating large steel plates.

Whereas with mild steel a scaling allowance has to be made, coupled with the fact that the processes used may have resulted in thinning, a spokesman for the company stated that in practice it was found that neither of these allowances proved to be necessary with the aluminium alloy ends. He also stated that research work on the handling of special aluminium alloys is being intensified at the company's works at Greenwich.

I.C.I. to Make Isodecanol

Isodecanol has now been added to the sales range of carbonylation alcohols manufactured by the Heavy Organic Chemicals Division of I.C.I. on Tees-side. This is the first time that commercial quantities of isodecanol have been made in Britain. Isodecanol is a mixture of branched-chain primary alcohols which is esterified with phthalic anhydride to produce the plasticiser D.I.D.P. (di-isodecyl phthalate). This plasticiser gives enhanced properties to P.V.C. products such as electrical cable and calendered film. Amongst the important characteristics so far established for D.I.D.P., such as good low-temperature and electrical properties and low density, its high permanence on thermal ageing is particularly notable. In a typical composition containing 33% by weight of plasticiser, the loss in weight of D.I.D.P. on heating for 24 hr at 100°C was only 0.3% (D.O.P. 0.5%) and for 350 hr at 100°C only 3.8% (D.O.P. 17.4%). Other carbonylation alcohols made by I.C.I. are iso-octanol, "Alphanol" 79, nonanol, *n*-butanol and isobutyl alcohol.



View showing the aluminium alloy ends, the largest ever manufactured by G. A. Harvey and possibly the largest in the United Kingdom, being processed at the company's works at Greenwich.

News Briefs

G. A. Harvey & Co. (London) Ltd. have recently delivered the first of two 120-ft absorber columns to the South Eastern Gas Board's Isle of Grain works. They are part of the high-pressure oil gasification installation now being built by the Woodall-Duckham Co. Ltd. The absorbers, Class 1 welded throughout, designed for operation at a working pressure of about 20 atm. (300 psi), are for the high-pressure CO_2 removal plant. They weigh approximately 100 tons each and are 9 ft 4 in. overall diameter.

In order to keep abreast of the rapid progress in the plastics industry, Shell have established a new plastics research centre at Delft in the Netherlands.

The new Petrolite facilities at Kirkby Industrial Estates, near Liverpool, are rapidly progressing toward planned completion on January 1, 1960. The new facilities will manufacture the full line of oil industry chemicals as offered by demulsifiers, corrosion inhibitors, water flood additives, fuel additives and scale preventives. Petrolite's Kirkby works will be supervised by C. N. Stehr, managing director.

The British subsidiary of The Dow Chemical Co., of Midland, Michigan, U.S. — Dobeckmun Britain Ltd. — has changed its name to Dow Chemical Co. (U.K.) Ltd. and broadened the scope of its activities. The company will take over direct sales of Dow-produced packaging films such as Trycite (Dow polystyrene film), Saran Wrap (Dow vinyl-vinylidene chloride copolymer film and polyethylene film. It will also supervise the sales of Dow U.S.- and Canadian-produced industrial chemicals,

Mild steel casing consisting of two Shells. Bottom shell 7'5" diameter x 17', top shell 6'6" diameter x 17'

How many craftsmen in this picture?

The answer is—dozens! For our specialised fabrication work embodies so many varied engineering skills. Customers call us in where individual quality and precision are vital. We fabricate in mild and stainless steels, and also cast in iron and all ferrous alloys, to meet your most exacting specifications—whether for fabrications or foundry work.

WIDNES foundry and engineering co ltd

Approved by Lloyds for class II welding for pressure vessels

Lugsdale Road • Widnes • Lancashire • Telephone Widnes 2251/4 and 2889 • Telegrams 'Foundry Widnes'

W54

plastics and magnesium in the United Kingdom.

Aerox Ltd. have, as a result of the steadily increasing demand for aeration, filtration and fluidising equipment, acquired a much larger factory for their engineering section at Chalford, Stroud, in Gloucestershire. At the new factory they will manufacture all forms of equipment using porous ceramic filter elements. These elements will continue to be made at the main Hillington (Glasgow) plant. The floor space of the new factory is five times greater than that of the old.

Imperial Chemical Industries has agreed to the introduction of a 42-hour week—working hours at present are 44—for roughly 60,000 of its manual workers. The new agreement, it is hoped, will become operative on January 4.

At the autumn research meeting of the Institution of Gas Engineers, Sir Harold Smith, chairman of the Gas Council, emphasised that the gas industry is a progressive industry. He thought that its most urgent job was to get the price of gas down. There were many new processes working or likely to be introduced in the next few years. The Gas Council is planning a national grid; they are aiming at a few large high-pressure gasification or hydrogenation stations using cheaper, smaller, non-caking coal with the gas going into a high-pressure grid. This would produce no coke and require no road transport. They are also searching for sites suitable for underground pressure storage.

John Dore & Co. Ltd. have taken over additional premises off the High Road, Plaistow. The new facilities will be used for the fabrication of chemical and allied plant in stainless and other steels and ferrous alloys.

Chemical Engineering Wiltons Ltd., of Stockport, have concluded an agreement with Lurgi-Gesellschaft für Chemotechnik MBH, under which Wiltons can offer in the U.K. activated-carbon solvent-recovery plants employing Lurgi processes.

Shell Chemical Co. Ltd. announce that they are to establish a plastics laboratory at their Carrington works, near Manchester. The laboratory, expected to be completed by 1961, is to support Shell's manufacturing and marketing of polystyrene and high-density polyethylene and future polypropylene and low-density polyethylene production.

Firth Cleveland Pumps Ltd. announce that they are to manufacture under licence the range of self-priming, multi-stage centrifugal pumps designed and developed by Dickow-Pumpen of Wailkrayburg, West Germany. Firth Cleveland will sell the pumps in all parts of the world.

Combining to form the Instrumentation Division of A.E.I. Ltd. are Sunvic Controls and the Instrument and Meter, Scientific Apparatus and X-Ray Departments of Metropolitan-Vickers Electrical Co. Ltd. The change-over becomes effective on January 1, 1960. In line with the former announcement, Birlec Ltd. will change to A.E.I.-Birlec Ltd. on the same date.

John Thompson Ltd. have opened a branch office in Buenos Aires, Argentina.

Metering Pumps Ltd., of Ealing, have now introduced a plant hire scheme to enable experimental work in chemical dosing to be carried out and for the effectiveness of new processes to be tested. During hire, service will be carried out at no additional cost and Metering Pumps' technical advice service will be freely available to clients.

The British Ceca Co. Ltd. announce that they can supply Clarcel filter aids manufactured in France by their associated company, Societé C.E.C.A. The full range of Clarcel grades are now available for all applications from U.K. depots.

Dorr-Oliver's offices at Abford House, Wilton Road, S.W.1, and the Mechanical Engineering Division at 21-22 Manor Mount, Forest Hill, have moved to: Norfolk House, Wellesley Road, Croydon, Surrey. Tel.: Municipal 2488.

Contracts

An order has recently been received from the Denver Equipment Co. for three classifiers valued at £19,163 (to be installed at a copper mine in Haiti) by Head Wrightson Stockton Forge Ltd., a subsidiary of Head Wrightson & Co. Ltd., to handle 500 short tons per day of copper ore.

What are stated to be the largest all-plastics structures ever made are being fabricated by Cawley Plastics Ltd. for a large chemical manufacturer. The contract is for a number of process towers made from dual laminate material, the largest single structure being a process tower 36 ft high and 4 ft internal diameter. Chemical storage tanks with capacities of 6500 gal. are also included.

Ferranti Ltd. announce they have received orders for two general-purpose Pegasus electronic computers, the total value of which exceeds £150,000. The first will be installed at the Road Research Laboratory of the D.S.I.R., where it will be used for all aspects of road research calculations. The second has been ordered by The Steel Company of Wales. Intended primarily for research studies, the computer is likely to be installed in the Operational Research Department at the Abbey Works, Port Talbot, within the next few months.

William Boby & Co. Ltd. have received contracts for demineralisation plants from The British Thomson-Houston Co. Ltd. and the Treforest Chemical Co. The former is for treating boiler water and the latter for the purification of gelatin solution.

Work is to start shortly on a new plastics factory at Speke, Liverpool, for the Metal Box Co. Ltd. The £325,000 contract has just been awarded to Holland & Hannen and Cubitts (North West) Ltd. Due for completion in the autumn of 1960, the single-storey factory will measure 270 ft by 250 ft.

Simon-Carves (Australia) Pty. Ltd. have received from the Sulphide Corp. (a subsidiary of the Consolidated Zinc Corp.) a contract of over £A1 million for a lead and zinc sinter plant at Cockle Creek in New South Wales. The sinter plant will be designed in London by Huntington, Heberlein & Co. Ltd., a subsidiary of Simon-Carves Ltd., of Stockport, but the major part of the plant and equipment will be manufactured in Australia. The sinter plant, part of the Sulphide Corp.'s £A8 million zinc smelter project, is expected to be in operation by March, 1961.

People in the News

The Minister for Science has appointed **Prof. Charles Frederick Carter** to be a member of the Council for Scientific and Industrial Research. He takes the place of **Prof. E. A. G. Robinson**, who retires on completion of his period of service. Prof. Carter is Stanley Jevons Professor of Political Economy and Cobden Lecturer in the University of Manchester. He is 40 years old. The present constitution of the Research Council is: chairman: **Sir Harry Jephcott;** members: **Sir Eric Ashby; Prof. C. E. H. Bawn; Prof. P. M. S. Blackett; Prof. C. F. Carter; Mr. H. Douglass; Sir Walter Drummond; Sir Willis Jackson; Vice-Admiral Sir Frank Mason; Sir Harold Roxbee Cox; Dr. C. J. Smithells; Mr. L. T. Wright;** and secretary: **Sir Harry Melville.**

Dr. H. R. Ambler has been appointed to the newly created post of Scientific Adviser to the U.K. High Commissioner in New Delhi. In addition to advising the High Commissioner on scientific matters, he will be responsible for furthering the exchange of information between British and Indian scientists.

The appointment of **John C. Weyrich** as manager for Du Pont's Maydown Works at Londonderry, Northern Ireland, has been announced. The Maydown Works will manufacture neoprene synthetic rubber primarily for the British and Western European rubber industry.

Mr. H. P. Forder and **Mr A. R. Hay** have been appointed directors of The United Steel Co. Ltd.

Mr. F. A. Ross, Mr. W. G. A. Jenkins and **Mr. J. Higginbotham** have been appointed directors of Edgar Allen & Co. Ltd.

Mr. J. A. Hepworth has been appointed chief engineer of the North Thames Gas Board as from January 1, 1960. He succeeds **Dr. J. Burns**, who, on that date, becomes deputy chairman.

The following have been elected to fill vacancies on the Council of The Institute of Metals with effect from the annual general meeting on March 29, 1960: as President: **Sir Ronald Prain**, chairman and president, Rhodesian Selection Trust Ltd; As Vice-Presidents: **H. M. Finniston**, research manager, C. A. Parsons & Sons Ltd., and **H. W. Hignett**, assistant managing director, Henry Wiggin & Co. Ltd.; As Ordinary Members of Council: **R. W. K. Honeycombe**, Professor of Physical Metallurgy, University of Sheffield; **I. Jenkins**, chief metallurgist, Research Laboratories, The General Electric Co. Ltd.; **E. Robson**, managing director, The Manganese Bronze & Brass Co. Ltd.; **J. Salter**, director, The British Aluminium Co. Ltd.; and **C. Smith**, works director, James Booth & Co. Ltd.